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WEAR

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ON FUNDAMENTALS OF FRICTION, LUBRICATION, WEAR,
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EDITORIAL

Old Fronts and New Frontiers

War on Wear was the title of a lecture given by H. BLOK some time ago[1]. This theme of "making war" on serious economic losses that are due to excessive wear has lately been elaborated by specialists dealing with maintenance problems of industry[2,3]. There are two well-known fronts in this "war": Maintenance and Production.

The contribution of Science to better maintenance is essentially of an analytical nature, leading to detailed knowledge of the causes of wear in a particular case. The main weapon against excessive maintenance costs[3], however, has the characteristics of operational research.

Production costs seem to be a better incitement to applied research than maintenance. Considerable efforts are made to develop *new tools* (see Abstracts section of this volume) and to find optimum conditions for the use of conventional tools, as is shown by the contributions of TRENT (p. 487) and SOLAJA (*Wear*, *2* (1958/59) 40, 311). Another aspect is the renewed attention given to the ultimate mechanical properties of such complex solids as coal and rock (see book review, p. 501), which should lead to a clearer picture of the tool–workpiece interaction. LANCASTER AND ROWE (p. 428) give a careful analysis of certain drawing operations which leads them to the conclusion that conditioning of the metal surface is decisive for an efficient use of the lubricant. ROZEANU AND PREOTESCU propose (p. 456) a mathematical analysis for the evaluation of optimum oil viscosity for automobile engines. The results of their calculations seem to agree well with field experience in other countries. This paper should perhaps be read in conjunction with other studies of the same group (p. 485). Obviously an audacious attempt is made to embrace in one mental picture phenomena due to elementary processes in the wear of metals and everyday problems of production and maintenance of engines.

Helpful though it may be, analysis of an existing wear problem usually refers to a practical case. New developments – the achievement of what even yesterday was "impossible" – are stimulated much more by *deductions* and by predictions based on general laws. CROOK (p. 364), in his survey on work done at Aldermaston, has shown *inter alia* how the thickness of the lubricant film in boundary lubrication is controlled by the temperature of the surface. ARCHARD now carries the story one step further by calculating the "flash temperatures" of such surfaces.* After reducing mathematical complexities, ARCHARD calculates flash temperatures for three typical examples.

* Since the original paper by H. BLOK (1937), giving the theoretical analysis, is rather inaccessible an amended republication was intended. However, so much new material has been received for publication in *Wear* that Professor BLOK has decided to abandon this plan.

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For steel rubbing on steel the temperature limit for martensitic transformation is reached under certain conditions of speed and load. Surface hardening occurs and the predicted reduction in wear rate is confirmed by experiments. LING AND SAIBEL (*Wear*, 1 (1957/58) 80) calculated the other alternative: catastrophic wear leading to galling of titanium and of steel. The same theory was applied, but under quite different experimental conditions. Relative changes in the wear rate of metals are thus *predictable* for given conditions of load, speed and temperature if:

(i) the thermal properties of the metal(s), and

(ii) the possible phase transitions of the metal(s) are known.

The second example in ARCHARD's paper refers to the surface temperature of Perspex rubbed by steel, for a series of loads and speeds of rubbing. Transition from the hard to the soft, visco-elastic state of Perspex is predicted from these computations — and confirmed by experiments — for speeds as low as 10 cm/sec, if only the loads are high enough.

This rather surprising consequence of flash temperatures, due to the low heat conductivity of plastics, may give a clue to the causes of complications in some of the surface friction experiments reported by BUECHE AND FLOM (p. 168). Some time ago SMEKAL[4] suggested that the amount of damage to surfaces due to scratching with a very fine needle depends on molecular flaws in solids. It will be necessary — at least with respect to non-metals — to reconsider this interpretation and take the contribution of thermal effects into account.

The third case dealt with by ARCHARD is that of flash temperatures within an oil film. It is shown that a temperature gradient in the film, studied experimentally by CROOK (p. 387), must exist and that the maximum temperature within the film will exceed many times that of the metallic surface. An immediate prediction of practical consequence is that the limiting temperature of thermal stability of an oil can be reached in boundary lubrication, while the machine parts are running apparently cool. Thus the old "front" of lubrication and maintenance is transformed into a new frontier, that of high transient temperatures. It is realized in industry that because of their thermal limitations application of conventional lubricants will decrease in the not too distant future. Their successors are already on the horizon: lamellar solids. Now that the lubricating action of graphite and MoS_2, placed at our disposal by Nature, is better understood, the search for "man-made" solid lubricants is on. Organic chemists may regain lost ground by creating plate-like "two-dimensional" condensed ring systems of higher thermal stability than that of available greases. Phthalocyanines already give promising results (p. 194).

At high Mach numbers frictional heating of solids by gases limits the operational possibilities of space flight. Calculations on the influence of, for example, sand paper roughness on the skin friction of solid surfaces (see Abstracts section) proceed along similar lines of thought as in the theory of flash temperatures. This *thermal wear*, an addition to the list of major wear types (*Wear*, 1 (1957/58) 120), forms a new barrier for the technology and economy of our time, the space age. To overcome this barrier, thermal transitions of metals and of (flame-sprayed and reinforced)

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ceramic coatings have had to be measured and taken into account, much in the same way as in the prediction of galling. Deduction was possible starting from existing physical principles. However, temporary protection against thermal wear has been achieved by a rather revolutionary approach: a gas layer produced from organic solids. This is a case of *transient gas lubrication*, where the elegance of theoretical and of laboratory concepts is surpassed only by their technical success.

G. SALOMON

Delft, September 1959

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FILM GEOMETRY EFFECTS IN HYDRODYNAMIC GEAR LUBRICATION

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SUMMARY

The involute gear lubrication problem is analyzed with a first order correction made to the conventional parabolic arc film thickness equation to account for the actual tooth shape and the squeeze action resulting from its time dependence. Both internal and external gears are considered for one pressure angle and a range of gear ratios. The film geometry effect is found to be greater for internal than for external gears and to decrease as the minimum film thickness decreases. Conclusions are drawn concerning the importance of the correction.

INTRODUCTION

In the analytical treatments of the hydrodynamic gear lubrication problem appearing in the literature since the pioneering work of MARTIN[1], it has been customary to approximate the geometry of the clearance between mating gear teeth by that of the clearance between rotating circular disks. The profiles of the circular disks are in turn approximated by parabolic arcs in the vicinity of contact when solutions to the equation governing the pressure distribution in the lubricant are attempted. Such solutions are readily obtained if viscosity variations, surface roughness, and surface deformations are neglected and often serve as the starting point for more advanced theories which attempt to take these complicating effects into account.

The author knows of no published investigation of the error inherent in this geometrical approximation. This error exists for two reasons. Gear teeth profiles are not parabolic arcs (or even circular arcs for that matter) and they change with time. This latter effect gives rise to a squeeze action not present in the clearance between circular disks and it is known that a squeeze action can have an important effect on the film pressure.

It is the purpose of this paper to present an investigation into the effect of the error in the geometrical approximation on the load capacity of the gear teeth. This investigation seeks to answer the question as to whether film geometry effects are important in the gear lubrication problem. Involute spur gears with a pressure angle of 20 degrees are considered and various gear ratios including internal gearing. The load capacity of the mating teeth is determined when the point of contact is at the pitch point.

The solution presented here is in the nature of a first correction to the conven-

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tional parabolic arc theory. An exact solution for the geometry effect would be extremely difficult to obtain and it is felt that for the purpose in mind the extra labor would not be justifiable.

NOTATION

r_1	pitch circle radius of driver gear
r_2	pitch circle radius of driven gear
φ	pressure angle
ω_1	angular velocity of driver gear
ω_2	angular velocity of driven gear
θ_1, θ_2	angles defining location of point of contact
ε	parametric angle in involute equations
x, y	cartesian coordinate system through point of contact
X	dimensionless x-coordinate, eqn. (16)
h	film thickness between mating teeth
h_0	minimum film thickness
H	dimensionless film thickness, eqn. (17)
b	dimensionless minimum film thickness, eqn. (20)
p	lubricant pressure
P	dimensionless lubricant pressure, eqn. (18)
μ	lubricant viscosity
U_1, U_2	gear tooth surface velocities
α	gear ratio (small to large)
β	auxiliary dimensionless parameter, eqn. (19)
w	load capacity per unit tooth width
W	dimensionless load capacity, eqn. (21)

THE FILM THICKNESS EQUATION

In this section an expression will be developed for the film thickness between mating involute gear teeth. The gear geometry under investigation is as shown in Fig. 1. The pitch circle radius of the driver gear is r_1 and that of the driven gear is r_2. The pressure angle is φ and the driver angular velocity is ω_1. The angles θ_1 and θ_2 define the location of the point of contact between two mating teeth along the line of action. Observe that when the point of contact is at the pitch point, both θ_1 and θ_2 are equal to tan φ.

We will now develop an expression for the profile shape of a tooth on the driver gear when its point of contact with the driven gear is at a location along the line of action defined by θ_1. A cartesian coordinate system is oriented as shown in Fig. 1, with the x-axis tangent to the tooth surface at the point of contact. Recalling the definition of an involute as the curve drawn by the end of a string which is kept taut

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while being unwound from a circle (in this case the base circle) we can readily derive the following equations for the tooth shape in terms of the parametric angle ε.

$$y = r_1 \cos \varphi \{\theta_1 + \sin \varepsilon - (\theta_1 + \varepsilon) \cos \varepsilon\} \tag{1}$$

$$x = r_1 \cos \varphi \{(\theta_1 + \varepsilon) \sin \varepsilon - (1 - \cos \varepsilon)\} \tag{2}$$

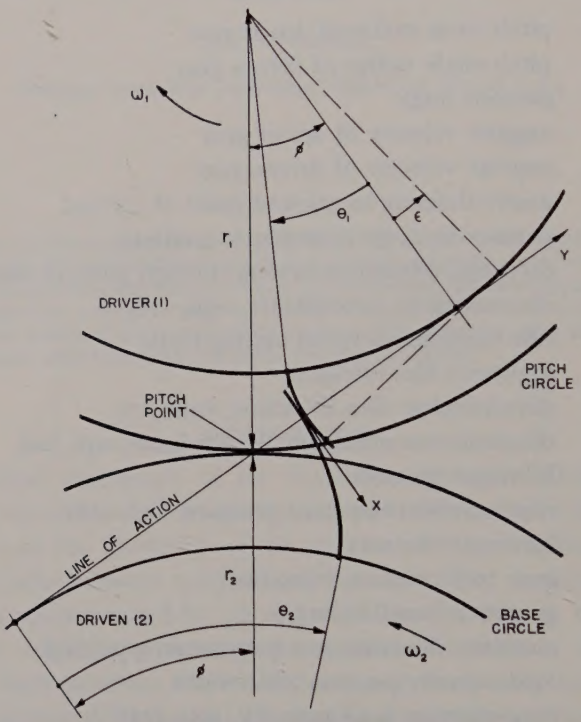

Fig. 1. Configuration of mating gear teeth.

We must now eliminate ε. This can be done explicitly if we realize that only small values of ε are of interest since the film pressure is significant only very near the point of contact. Replacing the trigonometric functions in eqns. (1) and (2) by their series expansions and neglecting terms of order ε^2 or smaller compared to unity there results.

$$y = r_1 \cos \varphi \left(\frac{\theta_1 \varepsilon^2}{2} + \frac{\varepsilon^3}{3}\right) \tag{3}$$

$$x = r_1 \cos \varphi \left(\theta_1 \varepsilon + \frac{\varepsilon^2}{2}\right) \tag{4}$$

Solving eqn. (4) for ε we obtain

$$\varepsilon = -\theta_1 \pm \sqrt{\theta_1^2 + \frac{2x}{r_1 \cos \varphi}} \tag{5}$$

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The minus sign can be disregarded for physical reasons. Expanding eqn. (5) to the same order of approximation as eqns. (3) and (4) we obtain

$$\varepsilon = \frac{x}{\theta_1 r_1 \cos\varphi}\left(1 - \frac{x}{2\,\theta_1^2 r_1 \cos\varphi}\right) \tag{6}$$

Substituting eqn. (6) into eqn. (3) there results

$$y = \frac{x^2}{2\,r_1\,\theta_1 \cos\varphi}\left(1 - \frac{x}{3\,r_1\,\theta_1^2 \cos\varphi}\right) \tag{7}$$

The parabolic arc theory uses only the first term of this expression, thus the second term can be regarded as a first order correction.

The film thickness between the mating teeth can now be expressed by

$$\begin{aligned} h = h_0 &+ \frac{x^2}{2\,r_1\,\theta_1 \cos\varphi}\left(1 - \frac{x}{3\,r_1\,\theta_1^2 \cos\varphi}\right) \\ &+ \frac{x^2}{2\,r_2\,\theta_2 \cos\varphi}\left(1 + \frac{x}{3\,r_2\,\theta_2^2 \cos\varphi}\right) \end{aligned} \tag{8}$$

where the contribution of the driven tooth has been derived from eqn. (7) by replacing r_1 with r_2 and reversing the sign on x. The reason for this procedure is easily seen by reference to Fig. 1. That the film thickness is time-dependent results from the fact that the angles θ_1 and θ_2 depend on time in accordance with the relations

$$\frac{d\theta_1}{dt} = \omega_1 \tag{9}$$

$$\frac{d\theta_2}{dt} = -\frac{r_1}{r_2}\,\omega_1 \tag{10}$$

We will assume h_0 to be independent of time.

THE PRESSURE EQUATION

In this section the appropriate pressure equation will be introduced and its solution presented for the film thickness variation derived above. The pressure equation governing this problem is the so-called generalized Reynolds equation (2) which allows for a time-dependent film thickness. This equation assumes the form

$$\frac{\partial}{\partial x}\left(h^3 \frac{\partial p}{\partial x}\right) = 12\mu\left(\frac{\partial h}{\partial t} + \frac{U_1 + U_2}{2}\frac{\partial h}{\partial x}\right) \tag{11}$$

where p is the lubricant pressure, μ its viscosity, and U_1 and U_2 are surface velocities of the driver and driven teeth respectively in the x-direction. To the order of approximation already employed in this problem it can be shown that at the pitch point

$$U_1 = U_2 = \omega_1 r_1 \sin\varphi \tag{12}$$

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The partial derivatives of h required by eqn. (11) can be obtained from eqn. (8). Evaluating them at the pitch point and substituting the resulting expressions into eqn. (11) we obtain

$$\frac{\mathrm{d}}{\mathrm{d}x}\left(h^3\frac{\mathrm{d}p}{\mathrm{d}x}\right) = -12\,\mu\omega_1\,x\left((1+\alpha) - (1+\alpha^3)\frac{x^2}{2\,r_1^2\sin^2\varphi\tan^2\varphi}\right) \tag{13}$$

where α is written for r_1/r_2.

In dimensionless form eqns. (8) and (13) can be written

$$H = 1 + (1+\alpha)\,X^2\left(1 \pm \frac{\sqrt{2}}{3}\beta\,(1-\alpha)\,X\right) \tag{14}$$

and

$$\frac{\mathrm{d}}{\mathrm{d}X}\left(H^3\frac{\mathrm{d}P}{\mathrm{d}X}\right) = -24\,X\left\{(1+\alpha) - (1+\alpha^3)\,\beta^2X^2\right\} \tag{15}$$

respectively, where

$$X = \frac{x}{\sqrt{2r\,h_0\sin\varphi}} \tag{16}$$

$$H = \frac{h}{h_0} \tag{17}$$

$$P = \frac{p}{\dfrac{\mu\omega r\sin\varphi}{h_0}\sqrt{\dfrac{2\,r\sin\varphi}{h_0}}} \tag{18}$$

and

$$\beta = \frac{1}{\tan\varphi}\sqrt{b} \tag{19}$$

with

$$b = \frac{h_0}{r\sin\varphi} \tag{20}$$

The quantity α is now redefined as the ratio of the smaller pitch circle radius to the larger. The smaller gear will be referred to in what follows as the pinion. Note that no subscript has been placed on r and ω in these dimensionless parameters and also that there is an alternate sign in the expression for H. This was done intentionally so that the reference for these parameters could be either the driver gear (subscript 1) or the driven gear (subscript 2), the choice to be made as follows. If the pinion is driving, then the reference is to the driver gear (ω and r take on the subscript 1) and the plus sign applies in eqn. (14). If the pinion is driven, then the reference is to the driven gear (ω and r take on the subscript 2) and the minus sign applies in eqn. (14). In other words the reference is always to the smaller gear. Because of

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the alternate sign possibility in the H equation we see that the load capacity of mating gear teeth depends on which of the two gears is driving.

Equations (14) and (15) describe the simple parabolic arc theory when b is set equal to zero. Thus b is a measure of the error inherent in the simple theory due to its geometrical approximation. We see that in the simple theory it does not matter whether the pinion is driving or driven.

The boundary conditions on the pressure equation are the usual Reynolds conditions; namely, that P approaches zero as X approaches infinity and that both P and its gradient vanish at the same (negative) value of X. Equation (15) with these boundary conditions and H given by eqn. (14) completely describes the pressure distribution in the lubricant film between gear teeth. Once P is obtained as a function of X by integration of eqn. (15) it must be integrated again over the film extent to yield the load capacity. The dimensionless load capacity will be defined by

$$W = \frac{w\, h_0}{2\, \mu\omega\, r^2 \sin^2 \varphi} \tag{21}$$

where w is the dimensional load capacity per unit width of gear tooth.

SOLUTION

Owing to the complexity of eqn. (15) the required integrations were performed numerically on an IBM-650 digital computer. The Runge-Kutta technique was used to determine P and the trapezoidal rule to determine W from P.

Results were obtained for a pressure angle of 20 degrees, three values of b (0, 10^{-4}, and $9 \cdot 10^{-4}$) and a range of values of the gear ratio α. The results are presented for the dimensionless maximum pressure (P_{max}) in Fig. 2, and for the dimensionless load capacity (W) in Fig. 3. A word of explanation is necessary about the abscissa in these two figures. The gear ratio α always means the ratio of the small pitch circle radius to the large. Values of α between 0 and 1 indicate external gearing (as shown in Fig. 1), and it can be demonstrated that negative values of α correspond to internal gearing. A zero value of α indicates, of course, a rack and pinion.

DISCUSSION AND CONCLUSIONS

From the results shown in Fig. 2 and 3 we see that for external gears of the same size ($\alpha = 1$) the first order correction to the simple theory is zero. If the gears differ in size, there is a correction and as the size difference grows the correction grows reaching its maximum value for a rack and pinion. For internal gearing the correction is greater than for external gearing, increasing as the gears approach each other in size. The correction increases the maximum pressure and the load capacity if the pinion is driving and decreases them if the pinion is driven. We also see that the correction increases in magnitude as the dimensionless minimum film thickness (b) increases. Recall that the results of the simple parabolic arc theory are obtained by setting b equal to zero.

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Since b will decrease as the dimensional load increases, we have shown that the film geometry correction decreases as the load on the gear teeth increases. This behavior is opposite to that of the other effects complicating the gear problem (viscosity variation, surface deformation, etc.) which all tend to increase along with

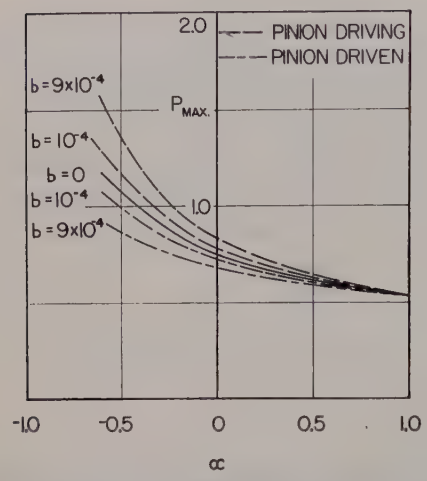

Fig. 2. Maximum pressure versus gear ratio.

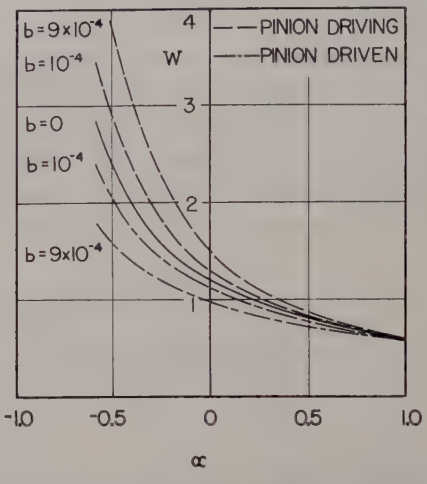

Fig. 3. Load capacity versus gear ratio.

the load. It is this condition of high tooth loading which is of current interest and since under this circumstance b would most likely be appreciable less than 10^{-4} inches we can probably say that as a first approximation in practical gear lubrication problems the film geometry effect can be neglected in comparison with the other complicating effects except possibly for internal gears of nearly the same size. It appears that the parabolic arc theory is never really valid since for light loads the film geometry effect is significant and for heavy loads the other complicating effects become serious.

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THE EFFECT OF THE ANISOTROPY OF THE BASE ON ROLLING FRICTION*

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SUMMARY

The subject of this paper is the experimental determination of the dependence of the rolling resistance on the direction of motion of the cylinder in the case of an anisotropic base of wood and NaCl monocrystals.

Experiment shows that a cylinder rolling on a horizontal plane diminishes its speed and comes to rest after some time. This can be described phenomenologically by introducing the notion of rolling "friction", or better still, of rolling resistance, as a force directed oppositely to the velocity of the cylinder axis. It has already been shown by COULOMB that this resistance T is directly proportional to the force N pressing the cylinder to the base, and inversely proportional to the cylinder radius R.

$$T = k\frac{N}{R}$$

The proportionality coefficient k, called the coefficient of rolling friction, has the dimension of length; its value depends on the material of both bodies, the state of their surfaces, the rolling speed, and, in the case of anisotropic bodies, on the direction in which the cylinder is rolling.

In order to choose the experimental conditions properly it is useful to consider which physical phenomena contribute to the loss of kinetic energy of the rolling cylinder. We list below the processes involved.

1. Compression of the surface asperities and jumps of cylinder over them.
2. Consolidation (strengthening) of the material of the cylinder and base.
3. Abrasion and wear of the material.
4. Cohesion of the cylinder and the base: TOMLINSON[1] in his molecular theory of friction sees as the essence of rolling friction precisely the cohesive forces between the cylinder and the base.
5. Sliding friction; the rolling cylinder, pressed to the base, becomes deformed and

* Condensed by the author from a paper published (in English) in *Acta Physica Polonica*, 17 (1958) 83.

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deforms the base. The parts of the cylinder in the area of contact are compressed, and parts of the base are under tension. As the cylinder rolls the area of contact moves, and the parts of the base not lying on the central line and released from compressive stress slide over the surface of the cylinder; the parts of the base surface coming under the cylinder similarly slide over the surface of the cylinder. REYNOLDS[2] and FÖPPL[3] ascribed a decisive role in the dissipation of energy to sliding friction and calculated the relation between rolling resistance and the coefficient of sliding friction.

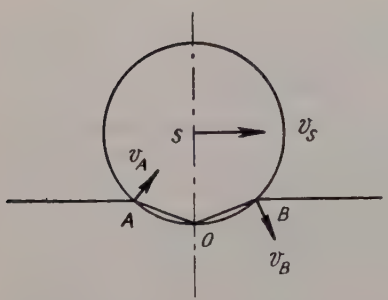

Fig. 1. Velocity of the points of the base in the area of contact with the cylinder.

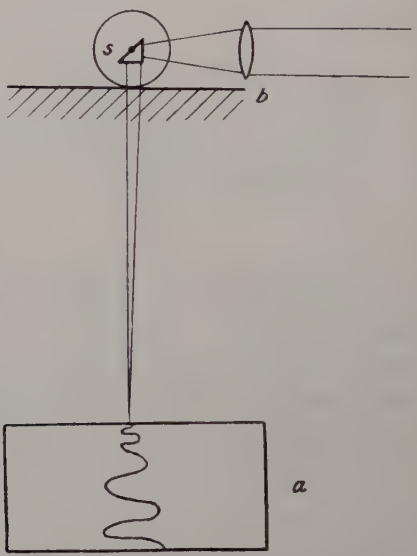

Fig. 2. Diagram of the arrangement for optical recording of the vibrations of the cylinder.

6. Elastic deformation of the base. Although at first glance it might seem that in the case of a perfectly elastic base the work done in compression by the front part of the cylinder is given up during the straightening out under the rear side, after considering, however, the velocities of the points of the cylinder in the area of contact (Fig. 1), we see that the velocity distribution, and therefore the pressure of the rolling cylinder on the base, is not symmetrical with respect to *OS* as in the case of a cylinder at rest. The rear part of the cylinder "moves away" from the base as the latter straightens out; the front part experiences inertial resistance from the base in addition to elastic resistance. This results from the moment of forces hindering the rotation. Another factor in the dissipation of energy is the imperfect elasticity of the cylinder and the base owing to the internal friction of solid bodies; even in the region of proportional deformations a body free from external forces does not give up all the work done on it.

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7. The production of elastic waves in the base. These waves carry away considerable energy. Thus, for example, the vibration of rails is observed at a distance of several km in front of and behind a moving train.

8. In the case of a large stress in the metal a plastic deformation takes place.

9. Finally, the rolling cylinder loses energy in overcoming the resistance of the air and the sound waves produced in it.

10. In the case of dielectrics, the rolling cylinder loses energy in overcoming the different electric polarization of the cylinder and base.

The percentage of each of these factors in the total resistance depends on the type of material, the pressure, the state of the surfaces, and the rolling speed. It seems that in the case of metal, at large pressures, the dominating role is played by plastic deformation; in the case of rubber, by internal friction. The phenomenon of cohesion can occur only with very clean surfaces.

Fig. 3.

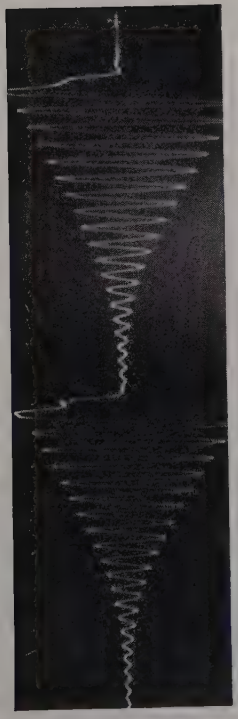

Fig. 4.

Fig. 5.

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The case of natural anisotropy was investigated for pine wood and sodium chloride crystals. For this purpose the very sensitive method of a vibrating cylinder with a lowered centre of gravity and resting on a horizontal base (Fig. 2) was employed. The centre of gravity of the cylinder was lowered by placing lead sheets on it eccentrically. A very small totally reflecting prism is located on the cylinder axis. It directs a beam of light onto a rotating cylinder covered with photosensitive paper. Lens *b* gives an image of a distant "point" source on the surface of the cylinder.

Fig. 6.

Fig. 7.

The cylinder displaced from the equilibrium position executes damped vibrations, successive amplitudes of which can be measured on the photogram. As a measure of the resistance we take for example the number of periods during which the amplitude is reduced to a value n times (*e.g.* 2) smaller than some initial value. This method is exceedingly sensitive, too sensitive to be used for investigating the rolling resistance in the case of the worked surface. It is seen from the photogram (Fig. 3) how the cylinder rolling perpendicularly to the furrows changes its amplitude in jumps and rebounds from their sides with a rocking motion of considerably shorter period.

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Figs. 4 and 5 represent photograms of the damping of the vibrations of a polished steel cylinder of 11.0 mm diameter loaded with lead disks to a mass of 1.7 kg on a pine wood base for the case (Fig. 4) with the axis parallel and (Fig. 5) with the axis perpendicular to the fibres of the wood. The difference in damping attains 50%.

A second example of directional dependence of rolling resistance in the case of natural anisotropy of the base is the rolling on the surface of a monocrystal. A natural NaCl monocrystal in the form of a cube with 5 cm edges was used. Since the natural faces of the cube have a layer-like structure one of the faces was polished. A vibrating steel cylinder of 10 mm diameter was placed first parallel to one edge of the polished face and then along its diagonal. Since the expected anisotropy could be caused primarily by the difference in the elastic coefficients of the crystal in each direction the cylinder was loaded on its axis with lead disks to a total vibrating mass of 4 kg in order to increase the elastic deformation. Fig. 6 shows the photogram of the vibrations of the cylinder with its axis parallel to the crystal edge; Fig. 7, with the axis making an angle of 45° with the edge. The effect is marked: in the case of the cylinder axis parallel to the crystal edge, the amplitude (double) drops after 30 vibrations from 33 mm to half the value; in the case where the cylinder axis lay along the diagonal — after 50 vibrations. The effect can be reproduced well at different places on the crystal and testifies to the great sensitivity of the method. It permits the discovery of elastic anisotropy in an optically isotropic crystal.

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A COMPARISON OF BOUNDARY LUBRICANTS UNDER LIGHT AND HEAVY LOADS

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SUMMARY

This paper describes an experimental study of the lubrication of steel bars during slow drawing under very heavy loads, (up to 10 tons), using dies of high entry-angle to discourage hydrodynamic contributions. The results are compared with those obtained using the same materials at light loads (1–4 kg). Soap and fatty acids are used to represent good boundary lubricants, while silicone oil is used as a standard poor lubricant and as a carrier for surface-active additives.

When the loads are increased from a few kilograms to a few tons the relative behaviour of the lubricants is the same, provided that the surface is not stretched more than a few per cent. If the loads are further increased the interfacial pressure (20–30 t.p.s.i.) is not greatly altered but the surface extends and the lubricant film breaks down, resulting in metallic pickup on the die. This can, however, be avoided by providing a regular pattern of pockets on the surface to trap the lubricant. A convenient way of doing this is to grit-blast the surfaces. When a reserve supply of lubricant is provided in this way in the vital region between die and specimen the lubrication does not break down. Under these conditions the boundary lubricating characteristics are again an important factor in the lubrication. The cross-sectional area of the bar may be reduced by up to nearly 60% in a single pass using soap as lubricant on a grit-blasted surface without pickup occurring. The limit is set not by pickup but by the strength of the drawn bar.

ZUSAMMENFASSUNG

EIN VERGLEICH VON GRENZSCHMIERMITTELN UNTER LEICHTER UND SCHWERER BELASTUNG

Die Schmierung von Stahlstäben beim langsamen Ziehen unter sehr schwerer Belastung (bis zu 10 Tonnen) wurde untersucht und zwar wurden Zugformen mit grossen Eintrittswinkel gebraucht, um den Anteil der hydrodynamischen Schmierung zurück zu drängen. Die Ergebnisse werden mit denen für dieselben Materialien unter niedriger Belastung (1–4 kg) verglichen. Seifen und Fettsäuren dienen als typische, gute Grenzschmiermittel, während Silikonöl als schlechtes Schmiermittel und als Träger für oberflächenaktive Zufügungen gebraucht wird.

Wird die Belastung von einigen Kilo's auf einige Tonnen erhöht so bleibt das relative Verhalten der Schmiermittel dasselbe, wenn die Oberfläche nicht mehr als einige Prozente gestreckt wird. Bei weiterer Druckerhöhung verändert der Zwischenflächendruck (30–45 kg/mm^2)) nicht wesentlich, jedoch wird die Oberfläche gedehnt, der Schmiermittelfilm bricht zusammen und die Form nimmt Metallteilchen auf. Dies kann jedoch verhindert werden durch für eine regelmässige Verteilung von Grübchen in der Oberfläche, die Schmiermittel festhalten, zu sorgen. Eine bequeme Methode um dies zu erreichen ist Sandstrahlen der Oberfläche. Wird so ein Schmiermittelvorrat in der entscheidenden Grenzzone zwischen Form und Material angelegt, so bleibt der Schmierfilm erhalten. Unter diesen Bedingungen werden die Eigenschaften des Grenzschmiermittels wieder wesentlich für die Schmierung. Mit Seife als Schmiermittel auf einer gesandstrahlten Oberfläche kann der Durchschnitt des Stabes in einem Arbeitsgang mit beinahe 60% reduziert werden ohne das Metallteilchen übertragen werden. Die Grenze wird nun durch die Zugstärke des Stabes und nicht durch die Metallübertragung bestimmt.

INTRODUCTION

There have been many studies of boundary lubrication under light loads (see, for example, ref.[1]). It is generally agreed that fatty acids and soaps form thin surface

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films which give low friction. However, when the loads are very heavy these lubricants appear to be ineffective. The films break down and metallic transfer occurs, causing severe damage and a marked increase in the friction[2].

The present experiments were performed under light loads (1–4 kg), and under heavy loads (1–10 T) giving reductions of area up to nearly 60% by drawing a square steel bar between two cylindrical dies.

EXPERIMENTAL

The bar-drawing apparatus was designed specifically to study the lubrication of metals being plastically deformed, and Fig. 1 shows the essential details of the

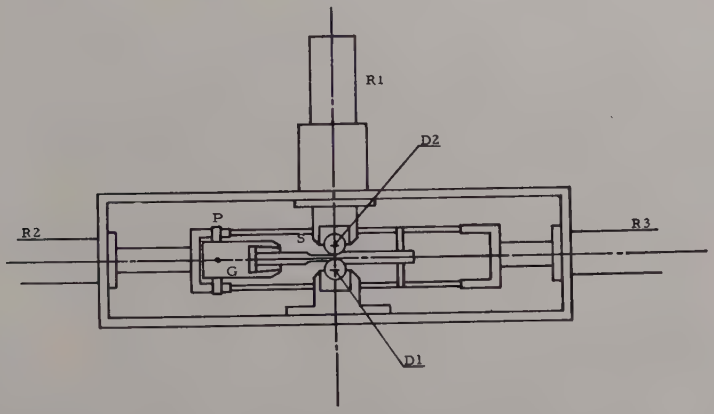

Fig. 1. Diagram of the bar-drawing apparatus. The die D1 is fixed to the case of the friction machine. D2 is another die housed at the end of a hydraulically operated ram R1. The specimen is secured at the left-hand end in the wedge grips of the shackle G, which is free to rotate about the axis of pin P. The specimen is drawn under approximately plane-strain conditions, by application of a force to the rigid carriage S from the ram R3.

working parts. The specimen is symmetrically indented by two dies which can have any desired profile, but for these experiments cylindrical dies were used, to discourage hydrodynamic effects. A hand-operated jack R1, can supply forces up to 20 tons for indentation.

The carriage through which the drawing forces are transmitted is a rigid assembly consisting of two thick end-plates connected by four 1-in. diam. bars of steel and fixed to the longitudinal rams R2, R3, which are actuated by a motor-driven pump so that reciprocating movement under pressure is possible. The specimen is held in position by jaws in a shackle similar to that of a tensile testing machine. Indentation and drawing forces are measured either by the pressure in the hydraulic rams, or by strain gauges attached to the die block and shackle.

Dies

Cylindrical dies were chosen for these experiments and were made of unplated Vibrac

steel which was hardened and tempered to give a hardness of 870 V.P.N. A smooth surface was obtained by hand grinding with carborundum papers to 4/0 (5 μ) grade before each drawing experiment. The dies were finally rinsed in ethyl alcohol immediately before use.

Specimens

Mild steel specimens of square section ($\frac{1}{2} \times \frac{1}{2}$ in.) were used. The analysis of the material, which was specially produced for these experiments, is:

C — 0.144	Si — 0.064	S — 0.037	P — 0.043
Mn — 0.520	Ni — 0.140	Cu — 0.220	Sn — 0.042

The steel was refined for 2 hours 5 minutes, in a basic open hearth furnace, and tapped at 1590°C. 3 in. square billets were taken from the middle of the ingot and rolled into $2\frac{3}{4} \times \frac{1}{2}$ in. flats in a 10 in. bar mill. These were then cut to size. All the specimens were annealed at 850°C, which produced an initial hardness of 110 V.P.N. Before lubricating each specimen, a standard surface finish was produced by hand grinding under a flood of water with silicon carbide papers to grade 600 (15 μ grit). They were given a final rinse in distilled water and dried in a stream of warm air. Hand contact was avoided during the final stages of grinding. This method produces grease-free surfaces but adsorbed films of air and water are not removed.

RESULTS

Light-load experiments

Numerous results have been obtained elsewhere with light-load apparatus[1]. For detailed comparison with the heavy-load drawing, however, some experiments were performed using the same annealed mild steel stock and a hard Vibrac slider. The loads ranged from 1–4 kg and various boundary lubricants were applied. The results are shown in Table I.

TABLE I

LIGHT-LOAD EXPERIMENTS (1–4 kg) VIBRAC ON MILD STEEL AND ON STAINLESS STEEL

Lubricant	μ
A. *Mild steel*	
Vacuum-cleaned steel	approx. 0.80 (at 2 kg)
Grease-free steel	0.2 — 0.25
Commercial soap	0.06
Sodium stearate	0.06
Paraffin oil	0.18
Oleic 5% in paraffin	0.12
Stearic 5% in paraffin	0.11
Lauric 5% in paraffin	0.13
Oleic acid	0.10
Stearic acid	0.09
Lauric acid	0.06
Silicone oil (100,000 cs)	approx. 0.5 (at 2 kg)
B. *Stainless steel*	
Grease-free	approx. 0.45 (at 2 kg)
Sodium stearate	0.08

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It will be seen that the fatty acids and soaps give good lubrication. It is not, however, possible to discriminate well between small additions of boundary lubricants to paraffin oil, since the paraffin itself shows a fairly low friction. However, it is known that silicone oil is a very poor lubricant, and so might be a more suitable carrier to allow the properties of the additives to show clearly. Some experiments with 100 cs silicone fluid showed that the friction rose steeply with load.

Typical values are:-

Load (kg):	0.5	1.0	1.5	2.0
Friction force (kg):	0.44	0.32	0.70	1.14
$\frac{\text{Friction}}{\text{Load}}$:	0.22	0.32	0.47	0.57

It was nevertheless found that the addition of 10% oleic acid to the silicone fluid gave a constant coefficient of friction $\mu = 0.12$, which is about the same value as was recorded for the mixture of oleic acid in paraffin, or indeed for the oleic acid alone.

Drawing experiments using boundary lubricants under heavy loads

The bar specimens were prepared in exactly the same way as the light-load specimens, by abrading on successive grades of emery under a flow of water, finishing with 600 grit (15 μ) emery paper. A wide variety of lubricants was used. Results are given for typical good and bad lubricants in Table II.

TABLE II

LUBRICATION OF SMOOTH MILD STEEL SURFACES UNDER HEAVY LOADS (1–5 tons)

	Reduction of area	*Die load (tons)*	*Drawing force (tons)*	μ*	*Remarks*
Grease-free steel	—	2.5	6.75	0.99	Broke
Soap (sodium stearate)	2%	2.25	0.65	0.09	Good surface
	9	3.75	1.52	0.11	Good surface
	18.7	4.75	2.39	0.12	Some pickup
	31.2	5.5	5.44	0.31	Bad pickup
Oleic acid	1	1.5	0.54	0.18	Smooth
	5.6	2.75	2.17	0.31	Pickup
	15.2	3.75	4.35	0.43	Bad pickup
Caproic acid	1	1.5	0.65	0.22	Smooth
	5.4	2.75	2.28	0.44	Pickup
	6.2	3.0	2.5	0.45	Pickup
Silicone oil	11.9	3.5	4.78	0.58	Very bad pickup and cracks
Silicone + 10% oleic	7.2	3.0	2.82	0.37	Scoring
	18.0	4.0	4.13	0.36	Bad pickup and scoring

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The generalised coefficient of friction μ^* is calculated by subtracting the frictionless drawing forces calculated on the basis of the GREEN AND HILL theory[3] from the actual force measured, and dividing this quantity by the die load. Several assumptions are involved, particularly that the drawing is performed under plane strain conditions, and a mean die angle is used for the circular dies. The result obtained is therefore valuable only for comparison of lubricants under similar conditions, and should not be compared numerically with the "true" coefficient of friction.

At the higher reductions the lubrication broke down and metallic transfer occurred. This seriously increased the friction and damaged the surfaces, as may be seen from Fig. 2. Drawing became practically impossible when the reduction of area exceeded about 30%. Eventually the drawn bar broke.

Fig. 2. Mild steel bars, finished by hand grinding, drawn between circular profile dies using oleic acid as lubricant. The pickup becomes progressively worse as the reduction is increased. (a) 1% reduction of area (b) 5.6% reduction of area (c) 15.2% reduction of area.

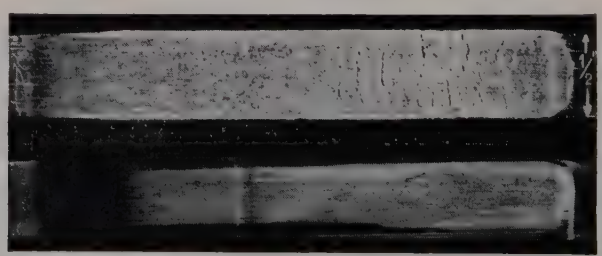

Fig. 3. (a) A bar drawn with silicone fluid shows severe cracking and pickup (12% reduction of area). (b) When 10% oleic acid is added to the silicone fluid the damage is much less severe (7.2 and 18% reduction of area).

Although at light loads a grease-free specimen shows only a moderate friction ($\mu = 0.25$), in the drawing experiments a grease-free bar broke immediately, although only 20% reduction was attempted.

When silicone oil is used, although the viscosity may be very high (100,000 cs)

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the lubrication is extremely bad. Even with only 10% reduction of area, the pickup is severe and both die and bar are seriously damaged (see Fig. 3). When 10% oleic acid is added the surface is greatly improved though scoring still occurs.

Thus it appears that the boundary-lubricating properties are important in determining the extent to which the bars may be drawn. Even with the best boundary lubricants, however, the draws are limited by pickup to a few percent.

The steel bars are work-hardened by drawing and the interfacial pressure is greater the heavier the reduction of area. However, this effect is much smaller than proportional increase in surface area. The yield stress, which determines the interfacial pressure, is not greatly altered (Table III). This suggests that the increase in pressure is not the major factor causing the breakdown of the lubricant.

TABLE III

MEAN YIELD STRESS OF MILD STEEL BARS DRAWN WITH DIFFERENT REDUCTIONS OF AREA

Load	*Reduction*	*Mean yield stress*
—	0	16.0 t.p.s.i.
3·75	9	21.0
4·75	19	24.6
5·5	31	29.0
5·75	50	33·5

Drawing experiments with trapped lubricants

The limitation in reduction of area imposed by pickup suggests that insufficient lubricant is present in the die–specimen interface. If a greater supply of lubricant can be ensured in this region, it may be possible to lubricate the whole of the increased surface adequately.

One means of ensuring this is to arrange for hydrodynamic lubrication, which relies on a low entry-angle of the die and a high drawing speed to force viscous lubricant into the die. When hydrodynamic conditions obtain, the boundary lubricating properties are of minor importance, since a thick continuous film is present between die and specimen. To examine boundary lubrication therefore we need to keep the speed low and the entry-angle large, as in the apparatus described in the EXPERIMENTAL section.

It is, however, possible to provide a local supply of lubricant to the stretching interface, independently of hydrodynamic behaviour, by trapping the lubricant in pockets on the surface.

Fig. 4 shows that if pockets are made in the surface, soap can easily be trapped. It is then highly compressed by the heavy forces acting on the bar (up to 30 t.p.s.i. for mild steel and 50–60 t.p.s.i. for stainless steel). The pocket is reduced in depth and elongated in the drawing direction and soap is forced out of the trailing end. Considerable quantities of soap are thus forced along the bar in the direction of oncoming

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metal. In this way each pocket can lubricate a considerable distance, at least ten times its own diameter (Fig. 5). In the immediate vicinity of the pocket the thickness of lubricant is sufficient to allow the metal surface to deform freely. This can be seen from the plastic roughening of the polycrystalline metal.

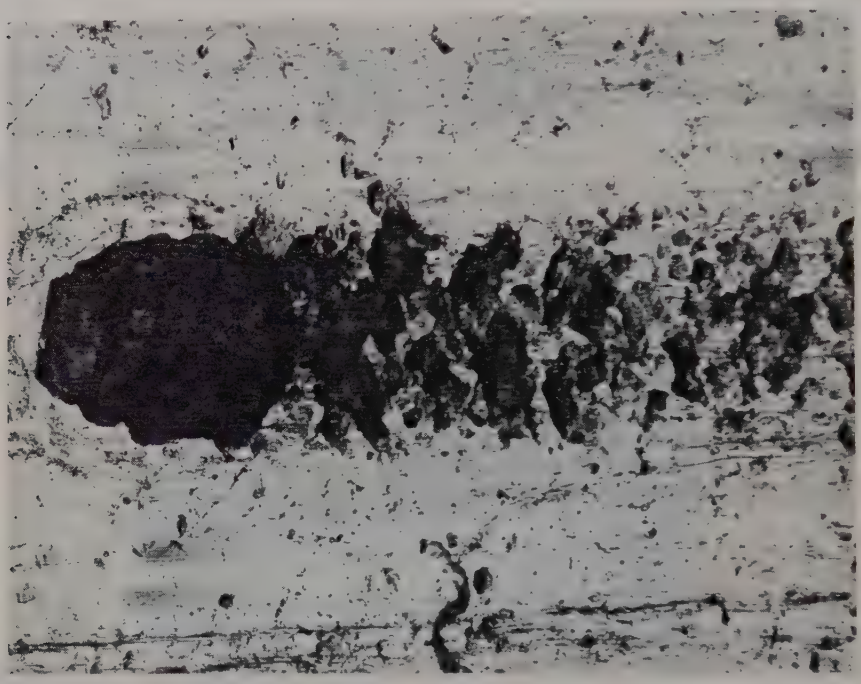

Fig. 4. When a spherical indentation is filled with sodium stearate, the lubricant is continuously fed out to protect the oncoming surface. The quantity of lubricant is sufficient to allow plastic roughening of the surface, for a considerable distance after the pocket. (× 40)

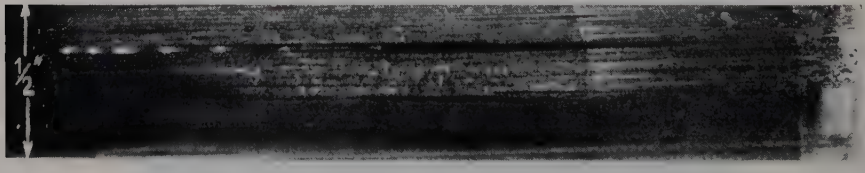

Fig. 5. A series of spherical indentations in line provides considerable protection of a soap-lubricated surface. In the line of the pockets there is complete absence of pickup but on either side of the line of pockets the smooth surface has been badly damaged.

The behaviour of these large isolated pockets can be reproduced on a small scale, and the lubrication can be greatly improved if the surfaces are covered with small pockets. Grit-blasting affords a convenient method of producing a uniform surface of this type.

It is found that with a good lubricant such as soap, grit-blasted stainless steel will draw to 57% reduction of area between hard steel dies without pickup. A similar

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specimen smoothed on 15 μ grit emery paper will pick up onto the die if reductions greater than 15% are attempted with the same soap. If the surface is highly polished pickup will occur with even lighter reductions. Once pickup starts it becomes rapidly worse and the friction increases (Table IV).

TABLE IV

LIMITING REDUCTIONS OF CROSS-SECTIONAL AREA OBTAINABLE WITH SOAP LUBRICATION

Surface condition	*Maximum reduction of area*		*Limitations*
	Stainless steel	*Mild steel*	
Grit-blasted	58%	56%	Tensile fracture without pickup
Abraded on 600 emery	17%	19%	Pickup and high friction
Polished	7%	13%	Pickup and high friction

As the drawing speed is reduced the results remain sensibly unchanged, as would be expected if the boundary properties predominate.

Some bars drawn with a viscous silicone oil show that the addition of small quantities of a boundary-active material is essential for good lubrication.

TABLE V

SILICONE FLUID WITH ADDITIVES, APPLIED TO GRIT-BLASTED MILD STEEL SURFACES

Lubricant	*Reduction*	*Die load*	*Drawing force*	μ^+	*Comments*
Silicone oil	22%	3.5	6.75	0.74	Heavy pickup, bad cracking
Silicone oil + 10%	8%	3	1.52	0.15	Smooth
oleic	27%	5	3.26	0.15	Smooth
Silicone oil + 10%	8%	3	1.52	0.15	Smooth
tri-cresyl phosphate	25%	5	3.04	0.19	Scored (no pickup)

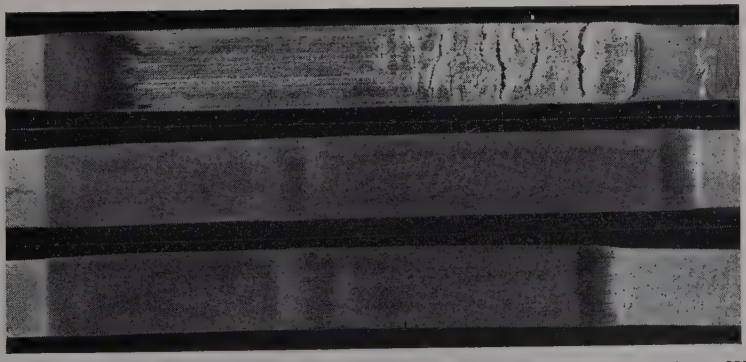

Fig. 6. By grit blasting, a uniform series of pockets can be made on the surface. When this is done the material can be drawn without pickup but the boundary properties of the lubricant are significant. (a) Poor boundary lubricant-silicone fluid (22% reduction). (b) 10% Oleic acid in silicone fluid (8.3% reduction, 27% reduction). (c) 10% Tricresyl phosphate in silicone fluid (8% reduction, 25% reduction).

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Fig. 6 shows that with the silicone oil alone the surface is very badly damaged but is quite smooth when the additives are present.

DISCUSSION

The preliminary light-load (1–4 kg) experiments show results in good conformity with previously published work[1]. Soap is the best of the lubricants used. The effectiveness of boundary additives can be seen when these are mixed with paraffin oil, but since the friction with paraffin oil alone is moderately low, this can be more strikingly seen with additives in a silicone oil.

Drawing under heavy loads (1–4 tons) shows that the relative merits of the boundary lubricants are not greatly altered, provided that the surfaces are not too much stretched. If, however, the reduction of cross-sectional area in the draw is too great, the lubricants break down on smooth bars prepared by grinding with 15 μ grit. This usually occurs by about 10% reduction of area, depending on the lubricant, but even the best lubricants allow pickup to occur at reductions in region of 20%. Heavier reductions seriously increase the extent of the damage to specimen and die although the yield stress, which determined the interfacial pressure, is not greatly increased.

If a reserve of lubricant, sufficient to cover the fresh extended surface, can be provided in the drawing zone this pickup can be avoided. The experiments show that small pockets on the surface can trap lubricant. This is then compressed and fed out under high pressure to the oncoming metal. Grit blasting has proved a very convenient method for producing pockets of this type. Heavy reductions of area, over 55%, for both mild and stainless steel, can then be taken with a good boundary lubricant without pickup occurring. The limit is then imposed by the strength of the drawn section. If heavier draws are attempted the specimen breaks, still without pickup.

To be effective on a grit-blasted surface the lubricant must have good boundary properties. Thus silicone oil, which is shown to be a very poor boundary lubricant, is also very poor when applied to grit-blasted surfaces for drawing. The addition of a few percent of a surface-active material greatly improves the lubrication of a silicone oil although the viscosity is at the same time reduced. Any hydrodynamic contribution will consequently be less for the oil with the additive.

Thus we conclude that there is a close correlation between light and heavy load behaviour provided the lubricant is compelled to enter the vital region. It has often been suggested that boundary lubricants are inadequate under severe drawing conditions (see, for example, ref. [4]). The present results suggest that this is not primarily due to the heavy pressures involved. It is generally accepted that high local pressures of the order of the yield stress will be developed in light-load experiments, and the average pressures in the heavy-load experiments reported in Table II are also about 20 t.p.s.i. Because the surfaces are not severely extended in either of these types of experiment the boundary lubricants are able to function satisfactorily. If, however, the surfaces are extended more than a few percent, the lubrication breaks down

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although the average pressure is increased only to about 25 t.p.s.i. On the other hand if sufficient lubricant is present to cover the new surface adequately, the relative performance of the boundary lubricants is again observed. Using grit-blasted surfaces, good lubrication can be obtained for drawing up to nearly 60% reduction of area without the help of hydrodynamic conditions.

ACKNOWLEDGEMENTS

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THE TEMPERATURE OF RUBBING SURFACES

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SUMMARY

A simple formulation of "flash temperature" theory is given. This treatment reduces the mathematical complexities and instead emphasises the relevant physical considerations. Simple graphical methods for deducing these temperatures are described. The use of the theory is illustrated by its application to some new experimental results on the breakdown of Perspex and the wear of steels. The extension of the theory to include the influence of surface (*e.g.* oxide) and lubricant films is outlined. The temperature in elasto-hydrodynamic films is considered; it is shown that the temperature difference which exists within such a film can be easily the largest transient temperature in the contact zone and may be several times greater than the surface flash temperature.

ZUSAMMENFASSUNG

DIE TEMPERATUR GLEITENDER OBERFLÄCHEN

Eine einfache Formulierung des Begriffes der Aufblitztemperatur wird gegeben. Die Behandlungsmethode reduziert die mathematischen Schwierigkeiten und legt den Schwerpunkt auf die wesentlichen physikalischen Betrachtungen. Einfache graphische Methoden für die Ableitung dieser Temperaturen werden beschrieben. Anwendungsbeispiele der Theorie werden an Hand einiger neuer experimenteller Resultate (Fressen von Perspex-Oberflächen durch Heisslaufen, Stahlverschleiss) erläutert. Die Erweiterung der Theorie auf Oberflächenhäute (z.B. Oxyde) und Schmierfilme wird kurz umrissen. Die Temperatur in elastohydrodynamischen Filmen wird untersucht; es zeigt sich, dass die in einem derartigen Film vorhandene Temperaturdifferenz leicht die höchste Übergangstemperatur in der Kontaktzone sein kann; sie kann die Aufblitztemperatur der Oberfläche um ein Mehrfaches übersteigen.

1. INTRODUCTION

Nearly all the energy dissipated by friction appears as heat and, at all but the slowest speeds of sliding, the temperature within the rubbing bodies is raised appreciably. The highest temperatures, which occur close to the areas of true contact at which the energy is dissipated, are of short duration and are sometimes called the flash temperatures.

These temperatures can obviously exercise a considerable influence on friction and wear and an estimate of their magnitude is often required. For example, it may be necessary to consider whether the surface temperature is consistent with the existence

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of certain oxides, with the production of phase changes in metals, or with surface melting. Because of their short duration and because they occur only over small regions, direct measurements of flash temperatures are difficult and a theoretical estimate is usually sought. A fairly complete theoretical analysis of the problem is available in the work of BLOK[1] and JAEGER[2].

The experimentalist requires a statement of this theory which omits its mathematical complexities and instead emphasises the physical considerations upon which the calculations are based. No complete treatment of this kind exists and Section 2 of the present paper attempts to meet the need. Simple graphical methods which can be used to determine the maximum possible flash temperature are then described. In Section 3 the use of the theory is illustrated by its application to some new experimental results on the breakdown of Perspex and the wear of steels. Finally in Section 4 it is shown that existing flash temperature theory is inadequate when applied to the condition of elasto-hydrodynamic lubrication; the temperature differences which then exist within the oil film may considerably exceed the rise in temperature of the surfaces.

2. THEORY

Nomenclature: –

A	area of contact
δA	individual areas in sub-divided contact (Fig. 1b)
a	radius of circular area of contact A
c	specific heat
E	Young's modulus
g	acceleration due to gravity
h	film thickness (Fig. 7)
J	mechanical equivalent of heat
K	thermal conductivity
$L = Va/2\varkappa$	dimensionless parameter
N	constant of proportionality (eqns. 9 and 14)
p_m	flow or yield pressure
Q	rate of heat supply from area A
$q = Q/A$	rate of heat supply per unit area
q_v	rate of heat supply per unit volume (Section 4)
R	undeformed radius of curvature of protuberance (Fig. 1)
t	duration of heat supply
V	velocity of heat source or speed of sliding
W	normal load
θ	rise in temperature above ambient value
θ_m	mean rise in temperature over contact area
θ_{max}	maximum rise in temperature within contact area
$\varkappa = K/\varrho c$	thermal diffusivity
μ	coefficient of friction
ϱ	density

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Subscripts B and C refer to bodies B and C (Fig. 1) and subscript 1 refers to the oil (Fig. 7).

2.1. General

The temperatures are calculated on the assumption that the heat is generated at the area of true contact (*i.e.* a single area of contact is regarded as a plane source of heat) and that this heat is conducted away into the bulk of the rubbing members. The theory firstly requires the solution of the equations for the flow of heat into each body. The derived surface temperatures are expressed in terms of the rate of supply of heat, the size and speed of the heat source, and the thermal properties of the material. Secondly, the proportion of the total heat flowing into each body is then determined by the criterion that the equations of heat flow for both bodies shall give the same average temperature over the contact area.

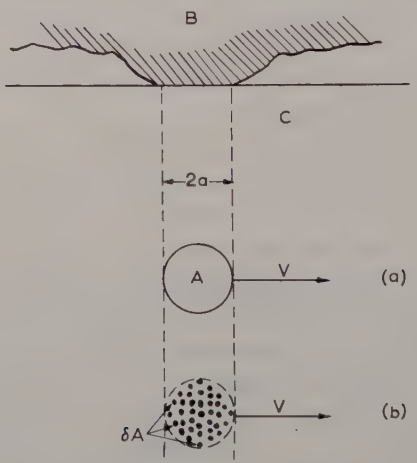

Fig. 1. The model. (a) Single contact area. (b) Contact area sub-divided.

The model of Fig. 1 will be used in this paper. A protuberance on the surface of body B forms a circular area of contact $A = \pi a^2$, which moves with a velocity V over the flat surface of body C. Body B therefore receives heat from a stationary heat source and body C from a moving heat source. The model of Fig. 1 is convenient for the present discussion but is not always applicable; in some examples (notably in the engagement of gear teeth) the area of contact moves over both surfaces. In such cases the theory for a moving heat source must be applied to both bodies.

2.2. The heat flow equations

Let Q_B be the quantity of heat supplied per second to body B and Q_C be the quantity of heat supplied per second to body C (Fig. 1). K_B and K_C are the thermal

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conductivities, c_B and c_C the specific heats and ϱ_B and ϱ_C the densities of the two bodies. The thermal diffusivities $\varkappa_B$ and $\varkappa_C$ are given by $\varkappa = K/\varrho c$. θ_m is the mean temperature of the contact area, taking the temperature of a point far removed from the contact as zero.

(a) *Stationary heat source.* For body B the heat is supplied to a fixed area and steady state conditions exist. The heat flow can be considered as a flow of thermal current through a thermal resistance and by comparison with the corresponding electrical problem[3,4] it is easy to show that

$$\theta_m = \frac{Q_B}{4aK_B} \tag{1}$$

(b) *Slow moving heat source.* Similar considerations will apply to body C when V is small; *i.e.* when V is so small that at each position of the contact there is ample time for the temperature distribution of a stationary contact to be established in body C. One then has, by analogy with eqn. (1)

$$\theta_m = \frac{Q_C}{4aK_C} \tag{2}$$

(c) *The speed criterion.* At higher speeds eqn. (2) will not apply because there is insufficient time for the temperature distribution of a stationary contact to be established in body C. The speed at which this occurs is related by JAEGER[2] to the dimensionless parameter*

$$L = Va/2\varkappa \tag{3}$$

To a reasonable approximation, eqn. (2) applies for $L < 0.1$, and for $L > 5$ the high speed eqn. (8) below, is applicable. In HOLM's[4] analysis faster moving contacts are regarded as being analogous to stationary contacts to which a source of heat is suddenly applied.

An alternative interpretation of eqn. (3) can be given in terms of the heat penetration into the bulk of the material. Consider the instantaneous generation of heat at the surface of C at a time $t = 0$. At a depth Z below the surface the maximum effect of this heat generation occurs after a time τ where[5]

$$\tau = Z^2/2\varkappa,$$

and when $Z = a$ (the radius of the contact area),

$$\tau = \tau_1 = a^2/2\varkappa. \tag{4}$$

The time τ_2 for the contact to move a distance a is

$$\tau_2 = a/V, \tag{5}$$

* JAEGER's analysis is for a square source instead of the circular source used here. The essential argument is not, however, affected.

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and therefore eqn. (3) may be rewritten as

$$L = \tau_1/\tau_2. \tag{6}$$

Thus, for large values of L the time taken for the heat to penetrate to a depth a (equal to the radius of the contact) is large compared with the time for which the heat source is applied.

(d) *Fast moving heat source.* The last paragraph shows that at high speeds ($L > 5$) the depth to which heat penetrates into body C during the time of contact is small compared with the dimensions of the contact. Therefore the sideways flow of heat can be neglected[1] and the problem can be treated as one of linear heat flow. In these conditions if heat is supplied at a constant rate* to the plane surface of a semi-infinite solid, the temperature θ of a point on the surface is[6]

$$\theta = \frac{2qt^{\frac{1}{2}}}{(\pi K \varrho c)^{\frac{1}{2}}}, \tag{7}$$

where q is the rate of supply of heat per unit area and t is the time for which it has been supplied. This equation can be applied to a fast moving contact by putting $q = Q_C/\pi a^2$ and by considering the effective value of t for any point within the contact. The temperature distribution over the area of a fast moving contact which has been deduced by JAEGER[2] is thus given a simple physical explanation. Equation (7) shows that the surface temperature is proportional to $t^{\frac{1}{2}}$; therefore in a square contact it is proportional to the square root of the distance from the leading edge of the contact. Using eqn. (7) and taking the average temperature, θ_m, over the circular area A of Fig. 1(a) one obtains

$$\theta_m = \frac{Q_C}{3.25\ K_C a}\left(\frac{1}{2L}\right)^{\frac{1}{2}} = \frac{0.31\ Q_C}{K_C a}\left(\frac{\varkappa_C}{Va}\right)^{\frac{1}{2}} \tag{8}$$

which, apart from a small change in the constant of proportionality, is identical with the equation given by JAEGER for a square source.

The results of the above theory are conveniently summarized in Fig. 2. The average temperature θ_m over the contact area is a function of the dimensionless parameter $L = Va/2\varkappa$. Then for a stationary heat source (graph A, Fig. 2) or for very low speeds ($L < 0.1$),

$$\theta_m = 0.5\ NL \tag{9a}$$

where $N = \pi q/\varrho c V$, q being the rate of heat supply per unit area. At higher speeds the temperature for a moving heat source (curve B, Fig. 2) falls below that given by eqn. (9a) and for moderately low speeds ($0.1 < L < 5$)

$$\theta_m = 0.5\ \alpha\ NL, \tag{9b}$$

where α is a function of L which can be obtained from Fig. 7 of JAEGER's[2] paper or

* Other types of heat supply distribution discussed by BLOK[1] and by JAEGER[2] can be treated in a similar manner.

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from the calculations of HOLM[4]. α ranges from about 0.85 at $L = 0.1$ to about 0.35 at $L = 5$. At still higher speeds ($L > 5$) eqn. (8) applies and

$$\theta_m = 0.435\, NL^{\frac{1}{2}} \tag{9c}$$

From eqns. (9a, b, c,) a smooth curve of θ_m/N as a function of L can be constructed (curve B, Fig. 2).

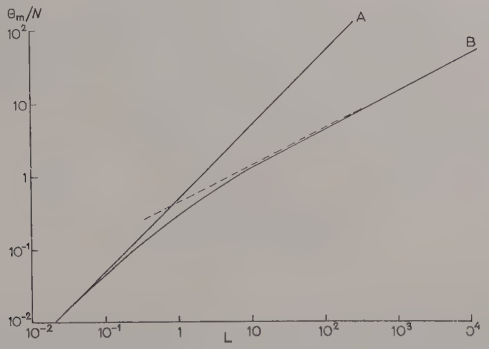

Fig. 2. Average temperature as a function of the dimensionless parameter L. A. Stationary heat source. B. Moving heat source.

To apply the results of Fig. 2 to a practical example the proportion of the total heat supplied to each of the two bodies must be taken into account. The author has found the following method convenient. First the flash temperature for each body is calculated on the assumption that *all* the heat is supplied to it. For example, for the model of Fig. 1 the temperature θ_B for body B is obtained from the stationary heat source theory (Fig. 2A) using a value L_B appropriate to the body B. The temperature θ_C for body C is obtained from the moving heat source theory (Fig. 2B) using a value L_C appropriate to the body C. Then the true temperature θ_m, taking into account the division of heat between B and C, is given by

$$\frac{1}{\theta_m} = \frac{1}{\theta_B} + \frac{1}{\theta_C} \tag{10}$$

2.3. Contacts whose size is determined by the applied load

Consider, for the sake of simplicity, the model of Fig. 1 with both bodies of the same material. The area of contact ($A = \pi a^2$) can be formed under a load by plastic or elastic deformation. If the deformation is plastic,

$$a = \left(\frac{W}{\pi p_m}\right)^{\frac{1}{2}} \tag{11}$$

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where p_m is the flow pressure or hardness of the material. If the deformation is elastic,

$$a = 1.1 \left(\frac{WR}{E}\right)^{\frac{1}{2}} \tag{12}$$

where R is the undeformed radius of curvature of the protuberances on body B (assumed spherical) and E is Young's modulus of the material.

The total rate of heat supply Q is $\mu WgV/J$, where μ is the coefficient of friction and J is the mechanical equivalent of heat. At slow speeds ($L < 0.1$) the heat supply is divided equally between both bodies and $Q_A = Q_B = \frac{1}{2}Q$. At high speeds ($L > 100$) practically all the heat is supplied to C (see eqn. (14d) below) and to a reasonable approximation $Q_C = Q$. Therefore at high and low speeds relatively simple formulae are obtained by combining eqns. (11) and (12) with eqns. (2) and (8) as follows:

With plastic deformation at low speeds ($L < 0.1$)

$$\theta_m = \frac{\mu g}{J} \frac{(\pi p_m)^{\frac{1}{2}}}{8K} W^{\frac{1}{2}} V \tag{13a}$$

With plastic deformation at high speeds ($L > 100$)

$$\theta_m = \frac{\mu g}{J} \frac{1}{3.25} \frac{(\pi p_m)^{\frac{3}{4}}}{(K\varrho c)^{\frac{1}{2}}} W^{\frac{1}{4}} V^{\frac{1}{2}} \tag{13b}$$

With elastic deformation at low speeds ($L < 0.1$)

$$\theta_m = \frac{\mu g}{J} \frac{1}{8.8K} \left(\frac{E}{R}\right)^{\frac{1}{2}} W^{\frac{1}{2}} V \tag{13c}$$

With elastic deformation at high speeds ($L > 100$)

$$\theta_m = \frac{\mu g}{J} \frac{1}{3.8} \left(\frac{E}{K\varrho cR}\right)^{\frac{1}{4}} W^{\frac{1}{4}} V^{\frac{1}{2}} \tag{13d}$$

Equations (13a-d) show how, under different experimental conditions, the average temperature at the contact area will vary in different ways with the load and the speed.

2.4. The maximum attainable flash temperature

In the interpretation of experiments the usual requirement is to know the maximum flash temperature which can be attained under the given experimental conditions. This occurs when the maximum possible load is concentrated at the smallest possible area, *i.e.* when the total applied load is borne by plastic deformation at a single area. Thus,

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consider the model of Fig. 1 with both bodies of the same material. The maximum flash temperature is obtained from eqns. (9) putting

$$N = \frac{\mu g}{J}\frac{\pi p_m}{\varrho c}, \qquad L = \frac{W^{\frac{1}{2}}V}{2\varkappa(\pi p_m)^{\frac{1}{2}}}$$

Then, at low speeds ($L < 0.1$) the heat supply is divided equally between B and C and

$$\theta_m = 0.25\, NL \tag{14a}$$

At moderately low speeds ($0.1 < L < 5$) eqn. (9b) applies, less than half the heat is supplied to body B, and

$$\theta_m = 0.25\, \beta\, NL \tag{14b}$$

where β ranges from a value of about 0.95 at $L = 0.1$ to about 0.5 at $L = 5$. At high speeds ($L < 100$) practically all the heat is applied to C and from eqn. (9c)

$$\theta_m = 0.345\, NL^{\frac{1}{2}} \tag{14c}$$

At somewhat lower speeds a proportion of the heat (appreciably less than unity) is supplied to C and θ_m falls below the value given by eqn. (14c). Thus at moderately high speeds ($5 < L < 100$)

$$\theta_m = 0.435\, \gamma\, NL^{\frac{1}{2}} \tag{14d}$$

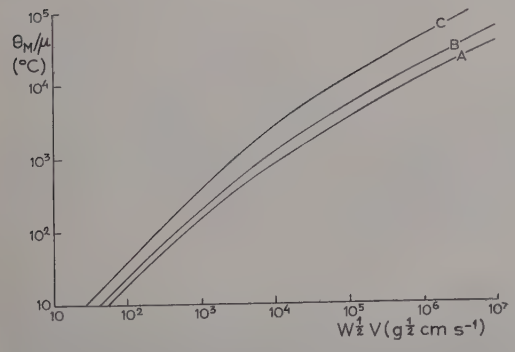

Fig. 3. Maximum attainable flash temperatures for sliding contacts of steel. A: V.P.N., 150. B: V.P.N., 250. C: V.P.N., 850.

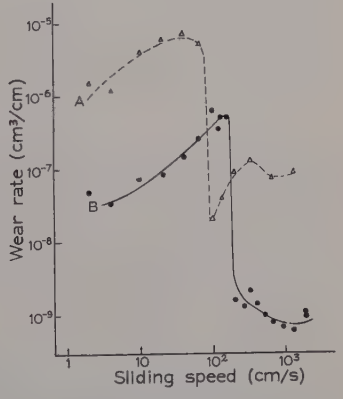

Fig. 4. Wear rates of 0.52% C steel as a function of speed. A: Load, 10 kg. B: Load, 1 kg.

where using eqn. (10) it can be shown that $\gamma = 1/[1 + 0.87\, L^{-\frac{1}{2}}]$. γ ranges from a value of 0.72 at $L = 5$ to 0.92 at $L = 100$.

Fig. 3 shows the application of these results to the important example of steel

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rubbing on steel. θ_m/μ is plotted as a function of $W^{\frac{1}{2}}V$. The position of the curve depends upon the hardness of the softer of the two rubbing members and the curves drawn are for steels of V.P.N. 150, 250 and 850. In deriving these curves the assumed values of ϱ, C and K are those of plain carbon steels; for high alloy steels account should be taken of their different thermal properties.

2.5. The effect of surface films

The influence of surface films may be of considerable importance. The most common example is that of a protective layer such as an oxide of low thermal conductivity. JAEGER[2] points out that such a film will raise the surface temperature and that to have any marked influence it must be thick compared with molecular dimensions. A complete analysis for film-covered surfaces would be quite complicated and it is sufficient here to indicate a few important conclusions.

A surface film of low thermal conductivity on the surfaces of the model of Fig. 1 will be considered. It will be assumed that the film thickness is small compared with the dimensions of the contact and that, apart from changes in thermal conduction, the experimental conditions (*e.g.* the coefficient of friction) are unchanged.

At low speeds of sliding it is convenient to use the electrical analogy. The thermal resistance of the contact now consists of the constriction resistance in the metal contact together with the film resistance in series with it. The heat supply (current), which is unaffected by the presence of the film, flows through these resistances and the temperature difference across any portion of the body is equivalent to the electrical voltage (IR) drop. Therefore, at low speeds of sliding the surface temperature is increased but the temperature drop across the constriction resistance, *i.e.* the substrate temperature, is unaffected by the presence of the film.

At high speeds of sliding the influence of the film is determined by its thickness compared with the depth of penetration of the heat (see paragraphs (c) and (d) in Section 2.2). For most practical speeds and values of the film thickness nearly all the heat penetrates to the substrate and its temperature is almost the same as that obtained without a film. Thus the effect of the film will be to raise the surface temperature and to lower or leave unaffected the temperature of the substrate. The substrate temperature cannot be increased by the presence of the film.

2.6. The sub-division of the contact region

In practice, a contact of the type shown in Fig. 1 may not be a continuous contact area A (Fig. 1a) but may instead be sub-divided into a number of smaller contact areas each δA (Fig. 1b). The temperature calculations can be applied to deduce the temperature distribution for the general contact region A or for the individual contact areas δA. It is important to consider the relative importance of the temperatures deduced in these two ways..

Consider the contact areas δA of Fig. 1b as being fairly closely packed. At slow speeds of sliding the thermal resistance consists of the constriction resistance of the

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whole region A in series with the sum (in parallel) of the constriction resistances of the areas δA. In the corresponding electrical example it has been shown experimentally and theoretically[7] that the aggregate resistance of a large number of small closely grouped constrictions, each δA, is very little greater than the resistance of one large constriction A; indeed for a single contact region between crossed cylinders the resistance between rough surfaces is nearly the same as that between smooth surfaces. Therefore, to a first approximation, the surface temperature of the model of Fig. 1b will be the same as that of Fig. 1a.

The above conclusions may be modified at higher speeds of sliding and for less closely packed contact areas. However, in general, the largest temperatures are those deduced for the whole contact region rather than those deduced for the smaller individual contact areas.

3. EXPERIMENTAL

3.1. The wear of steels

Steel surfaces, when rubbed under relatively severe conditions, exhibit structural changes and intense hardening of the surface layers; these changes can cause a marked decrease in the wear rate[8]. These effects have been attributed to the formation of martensite and it will now be shown that the local temperature rise required for the martensitic transformation (at least 700°C) is consistent with the flash temperature theory outlined above.

Fig. 4 shows the results of some wear measurements* with a pin and ring machine[9]. Both pins and rings were of a 0.52% carbon steel (V.P.N. 250). Graph A shows the wear rate of the pin as a function of the speed for a load of 10 kg. At low speeds severe wear occurred and the wear rate was high. At a speed of about 100 cm/sec the characteristic reduction in the wear rate associated with the martensitic transformation occurred. With a load of 1 kg (Fig. 4B) a similar behaviour was observed, the reduction in the wear rate occurring at a speed of about 200 cm/sec.

The maximum theoretical flash temperatures for the transition conditions of Fig. 4 can be obtained from Fig. 3B. A coefficient of friction of unity will be assumed (friction machine experiments show that during welding μ rises to between 0.5 and 1.5). At 100 cm/sec the full load of 10 kg produces a temperature rise of 1,200°C whilst a rise of 700°C is produced by a local load of 2.5 kg. At 200 cm/sec the full load of 1 kg produces a temperature rise of 900°C and a rise of 700°C is produced by a local load of 600 g. The theoretical flash temperatures are therefore more than sufficient to justify the assumption that the hard wear-resistant surface layers are formed as a result of frictional heating. A more exact comparison between theory and experiment is not possible because of uncertainties about the nature of the experimental conditions. For example, theory forecasts that the flash temperature depends upon the parameter $W^{\frac{1}{2}}V$; a tenfold reduction in the load might therefore be expected to cause the transition speed to increase by a factor of three. The experiments show that the transition speed shows only a twofold increase (Fig. 4). However, as pointed out by

* The author is indebted to Mr. A. J. PAYNE for his help with these experiments.

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WELSH[8] the reduction in wear is not associated with the first appearance of the martensitic transformation but with the formation of quantities of this material sufficient to arrest the progress of severe wear.

3.2. Thermal breakdown in the wear of high polymers

Experiments show that at light loads and moderate speeds thermoplastic materials such as Perspex and Polythene have a comparatively low wear rate. However, if the load or speed is increased a point is reached where there is a catastrophically large increase in the wear (Fig. 5) and this breakdown is attributed to surface melting of the polymer. The following experiments* were devised to determine the conditions for this thermal breakdown at a single contact region.

A pin and ring machine was used with the specimens oriented in the crossed-cylinders arrangement (Fig. 6). Contact between the specimens was therefore confined to a small elliptical region. The load W was applied and the machine was then run for

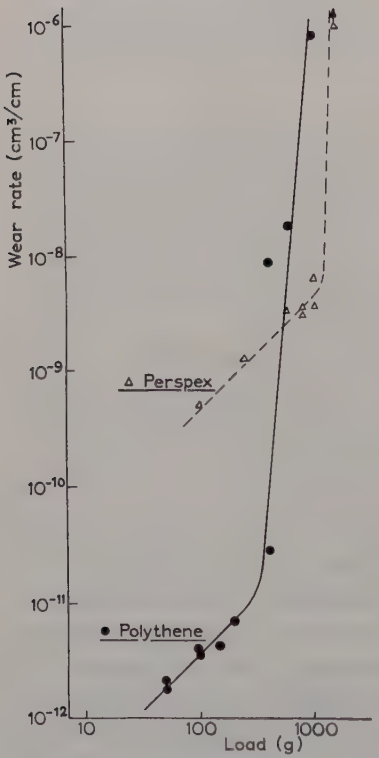

Fig. 5. The wear of polymers on hardened steel. Speed 200 c/sec.

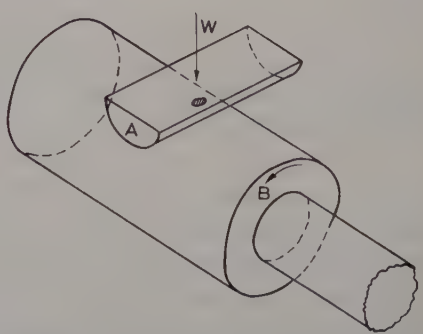

Fig. 6. Experimental arrangement for the investigation of thermal breakdown in the wear of Perspex. A: Semi-cylindrical pin of Perspex, diameter 1.25 cm. B: Rotating ring of Perspex, diameter 2.72 cm. W: Applied load.

* The author is indebted to Mr. M. E. BAKER for his help with these experiments.

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a short period (usually one minute). At light loads little or no damage could be detected after this period of running. In subsequent runs the load was increased in small steps until examination showed severe damage of the pin. In this way it was possible to determine, with an accuracy of about 10%, the load required to cause the onset of heavy wear. The appearance of the specimens supported the suggestion that this breakdown was caused by surface melting. With loads below the transition load damage was seldom evident, the most that was observed being a few fine scratches. With loads above the transition load it was obvious that considerable surface flow had occurred and usually there was a well-defined wear scar. Breakdown loads were measured for a number of speeds; they ranged from about 20 g at a speed of 356 cm/sec to about 3.5 kg at a speed of 11.4 cm/sec. The results are shown in Table I. The coefficients of friction listed are the average values at or near the breakdown load obtained in an independent series of experiments carried out under similar conditions.

The specimens were polished to a fine surface finish ($< 2\mu$in. C.L.A.) using metal polish and were carefully washed in petroleum ether before each run. This method of cleaning gave lower values of friction than vapour degreasing but resulted in fairly consistent friction measurements and breakdown loads.

TABLE I

THERMAL BREAKDOWN OF PERSPEX

Breakdown load (W), coefficient of friction (μ) and radius of contact area (a) in experiments at different speeds (V). The radius a is the geometric mean of the semi-axes of the ellipse (ratio of axes = 1.7 approximately).

V(cm/sec)	W(g)	μ	a (mm)	$\mu W^{\frac{1}{2}}V^{\frac{1}{2}}$
11.4	3250–3500	0.09	0.55	18.4
21.4	1400–1500	0.11	0.43	19.0
35.6	650–700	0.12	0.32	19.2
71.2	275–325	0.14	0.25	20.8
114	100–110	0.17	0.17	19.0
214	40–50	0.20*	0.13*	19.7
356	20–22	0.23*	0.10*	19.9

* Extrapolated from measurements at heavier loads.

Area of contact measurements (Table I) are, to a reasonable approximation, consistent with eqn. (12) which assumes that the deformation is elastic. Application of eqn. (3) showed that the values of L range from 280 at 11.4 cm/sec to 1600 at 356 cm/sec. Thus the high-speed equations apply and the temperature at the contact should be given by eqn. (13d), which implies that the surface temperature should be proportional to $\mu W^{\frac{1}{2}}V^{\frac{1}{2}}$. Therefore if breakdown occurs when the surface temperature reaches the softening point of the Perspex, $\mu W^{\frac{1}{2}}V^{\frac{1}{2}}$ should be constant for all the experiments. The last column of Table I shows $\mu W^{\frac{1}{2}}V^{\frac{1}{2}}$ was constant despite wide changes in the load and speed.

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The theoretical value of the maximum temperature in the contact can now be calculated. Equation (13d) gives the average temperature in a circular contact and this is increased by a factor of 1.14 to take account of the elliptical shape of the contact in the present experiments. The maximum temperature θ_{max} is 1.65 times the average temperature and we therefore write for these experiments

$$\theta_{max} = 0.50 \frac{g}{J} \left(\frac{E}{K\varrho cR}\right)^{\frac{1}{2}} \mu W^{\frac{1}{2}} V^{\frac{1}{2}}$$

The material constants in the above equation are as follows: $K = 4.7 \cdot 10^{-4}$ cal sec^{-1} cm^{-1} $°C^{-1}$ (measured in a Lee's disc experiment), $\varrho = 1.2$ g cm^{-3}, $c = 0.35$ cal g^{-1} $°C^{-1}$, $E/R = 2.7 \cdot 10^7$ g cm^{-1} (obtained from the area measurements). Thus using the average value of $\mu W^{\frac{1}{2}} V^{\frac{1}{2}}$ from Table I (19.4 $g^{\frac{1}{2}} cm^{\frac{1}{2}} sec^{-\frac{1}{2}}$) we obtain

$$\theta_{max} = 84°C$$

This is close to the softening point of Perspex, which is about 80°C above the ambient temperature. There is therefore good agreement between theory and experiment.

4. TEMPERATURES IN ELASTO-HYDRODYNAMIC CONDITIONS

Elasto-hydrodynamic conditions occur in heavily loaded bearing elements such as gears and ball or roller bearings where it is agreed that the oil film can be much thicker than that forecast by classical hydrodynamic theory. The surfaces deform under the high local pressures and it is known that the increase of lubricant viscosity with pressure plays a vital role in determining the value of the film thickness. The subject has been reviewed recently by BLOK[10].

The temperatures which exist in elasto-hydrodynamic films will now be discussed. The theory which is developed below will be illustrated by reference to a disc machine experiment described by CROOK[11,13]. Two steel discs of 3 in. (7.62 cm) diameter were run together with unequal peripheral speeds V_1 and V_2. The experimental conditions are illustrated in Fig. 7. At a load of $2 \cdot 10^8$ dynes/cm of face width the film thickness

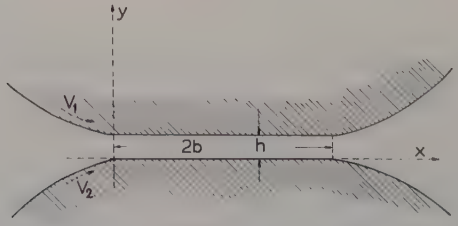

Fig. 7. The elasto-hydrodynamic film. In CROOK's disc machine experiment discussed in the text the details are as follows: Load per unit face width $W' = 2 \cdot 10^8$ dyne/cm, $2b = 4.2 \cdot 10^{-2}$ cm. $h = 1 \cdot 10^{-4}$ cm, $\mu = 0.04$, $V_1 = 596$ cm sec^{-1}, $V_2 = 920$ cm sec^{-1}, mean peripheral speed, $V = 758$ cm sec^{-1}. For the oil, $K_1 = 4 \cdot 10^{-4}$ cal cm^{-1} sec^{-1} $°C^{-1}$, $c_1 = 0.5$ cal g^{-1} $°C^{-1}$, $\varrho_1 = 0.8$ g cm^{-3}, $\varkappa_1 = 1 \cdot 10^{-3}$ cm^2 sec^{-1}.

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was shown to be about $1 \cdot 10^{-4}$ cm and the coefficient of friction was 0.04 (CROOK, private communication). In accordance with elasto-hydrodynamic theory it will be assumed that the deformation of the discs was Hertzian and consequently a parallel film (Fig. 7) of length $2b = 4.2 \cdot 10^{-2}$ cm was formed between the surfaces. Flash temperature theory, which assumes that the effect of the oil film can be neglected, gives a maximum surface temperature of 21°C. From simultaneous measurements of the electrical resistance and electrical capacity, CROOK estimated that the temperature of the oil passing between the discs was raised by about 200°C. This estimated temperature rise is so far in excess of the theoretical surface temperature that the applicability of the flash temperature theory to elasto-hydrodynamic conditions clearly requires reconsideration.

The heat is generated by viscous losses in a parallel slab of oil of thickness h and length $2b$ (Fig. 7). As a first approximation it will be assumed that this slab passes between the discs with their mean velocity $V = \frac{1}{2}(V_1 + V_2)$ and that the heat is generated uniformly throughout the slab. It is important to recall that in practice h is much smaller than $2b$ ($h = 1\,\mu$, $2b = 420\,\mu$ in our example). Since the film is bounded by material of much higher conductivity the heat flow in the oil is in the y direction. The problem is therefore one of linear heat flow.

Each element of the film has received heat for a period of time $t = x/V$, x being the distance of the element from the entry side. The analysis is therefore that of the temperature distribution in a slab of material in which a uniform supply of heat, q_v units per unit volume, is started at a time $t = 0$. The boundary conditions (Fig. 7) are, $\theta = 0$ at $t = 0$ for all values of y, and

$\theta = 0$ at $y = 0$ and $y = h$ for all values of t,

where θ is the temperature.

The solution of this problem[12] is

$$\theta = \frac{q_v}{2K_1}\left[hy - y^2 - \frac{8h^2}{\pi^3}\sum_{m=\infty}^{m=0}\frac{1}{(2m+1)^3}\left(\exp - \frac{\varkappa_1(2m+1)^2\pi^2 t}{h^2}\right)\sin\frac{(2m+1)\pi y}{h}\right] \quad (15)$$

where K_1 is the thermal conductivity of the oil, $\varkappa_1$ its thermal diffusivity, and y the distance measured from one face. On the median plane ($y = h/2$) this reduces to

$$\theta = \frac{q_v h^2}{8K_1}\left[1 - \frac{32}{\pi^3}\sum_{m=0}^{m=\infty}\frac{(-1)^m}{(2m+1)^3}\left(\exp - \frac{\varkappa_1(2m+1)^2\pi^2 t}{h^2}\right)\right] \quad (16)$$

It is clear that as t increases θ rises towards its steady state value θ_∞ and that a close approximation to steady state conditions exists when the summation term in eqn. (16) is small compared with unity. It can be shown from eqn. (16) that θ is 95% of θ_∞ when $t = 0.3\ h^2/\varkappa_1$. We will assume that the thermal conductivity of the oil

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K_1 is $4 \cdot 10^{-4}$ cal cm^{-1} sec^{-1} °C^{-1}, its specific heat c_1 is 0.5 cal g^{-1} °C^{-1} and its density $\varrho_1 = 0.8$ g cm^{-3}. The thermal diffusivity $\varkappa_1$ is therefore $1 \cdot 10^{-3}$ cm^2 sec^{-1}. Thus in the example of Fig. 7, where $h = 1 \cdot 10^{-4}$ cm, a good approximation to steady state conditions is established in $3 \cdot 10^{-6}$ sec. In this example the mean velocity of the oil passing through the parallel region is 758 cm sec^{-1} and each element of oil is therefore heated for a total time of $56 \cdot 10^{-6}$ sec. This time is large compared with the few microseconds required for the establishment of steady state conditions. Consequently steady state conditions will exist over nearly all the width $2b$ of the parallel film. The oil temperature on the median plane is therefore

$$\theta = q_v h^2 / 8K_1 \tag{17}$$

The rate of heat supply per unit volume q_v is given by

$$q_v = \frac{\mu W' (v_1 - v_2)}{2Jbh} \tag{18}$$

where J is the mechanical equivalent of heat and the other symbols are defined in Fig. 7. Thus, in our example,

$$q_v = 1.5 \cdot 10^7 \text{ cal sec}^{-1} \text{ cm}^{-3}$$

and from eqn. (17) the temperature rise on the median plane is 47°C. This temperature is attained under steady state conditions; this means that it is the temperature difference within the oil film required to ensure that all the heat generated is conducted to the surfaces. Under these conditions the surface temperatures will be close to those deduced from flash temperature theory since the only change introduced by the presence of the oil is in the division of the total heat supply between the two surfaces. Therefore, to a first approximation, the temperature on the median plane is the temperature of eqn. (17) plus the corresponding surface temperature calculated from the flash temperature formula. In Fig. 8 these temperatures are plotted for the example of Fig. 7.

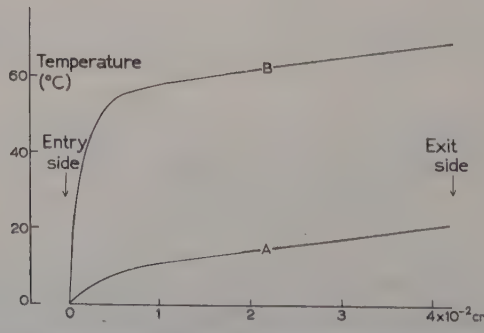

Fig. 8. Temperatures for the example of Fig. 7 assuming uniform heat dissipation. A: Surface temperature. B: Temperature at the centre of the oil film assuming uniform distribution of heat supply throughout the parallel film.

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The original assumption was that the heat, generated by viscous losses, was distributed uniformly throughout the parallel film. This is incorrect because wide variations in viscosity will exist. The appropriate modifications to the initial assumption will now be considered and their effect upon the calculations will be briefly discussed.

Firstly, because of the temperature gradient, the viscosity will be least along the median plane ($y = h/2$) and greatest at the surfaces ($y = 0$ and $y = h$). If an exaggerated view of the situation is taken and it is assumed that, compared with the oil in the median plane, the rest of the film is solid, then all the shearing and all the heat generation occurs on the median plane. The actual situation will be intermediate between this extreme and the uniform heat generation originally assumed. Along a line parallel to the y axis (Fig. 7) the shear stress is approximately constant and it is easily shown that under constant shear stress conditions the rate of shear strain and the rate of heat generation are inversely proportional to the viscosity. As an upper limit one can assume that all the heat is generated on the median plane. Since, as shown earlier, a steady state temperature distribution exists the calculation is elementary and the temperature rise on the median plane is

$$\theta = q_v h^2 / 4K_1$$

This is twice the value obtained previously (eqn. (17)) assuming a generation of heat uniformly distributed throughout the film.

Secondly, under elasto-hydrodynamic conditions wide variations in viscosity are to be expected because the viscosity increases with the pressure. Since the pressure distribution is approximately Hertzian the viscosity rises from a low value on the entry side, reaches a maximum at the centre, and falls towards the exit side. In shearing this parallel film of variable viscosity the greatest rate of heat generation is to be expected at the centre where the viscosity is greatest. Thus, along a line parallel to the x axis (Fig. 7) the rate of shear strain is approximately constant and under constant rate of shear strain conditions the shear stress and the rate of heat generation q_v are proportional to the viscosity. Obviously q_v will rise towards the centre, where the viscosity is greatest and will then fall towards the exit side. In the earlier discussion it was shown that a steady state temperature distribution was rapidly established within the film. Therefore the temperatures within the film will follow quite closely the changes in q_v.

The variations in q_v in the x direction will also influence the distribution of temperature over the surfaces[1] but the surface temperatures will respond more slowly to those variations in q_v. Thus, the main effect of variations in viscosity is to increase the maximum temperature within the oil film, to move this maximum towards the centre of the parallel film, and to exercise a smaller influence upon the surface temperatures.

A conservative estimate of the temperatures in the above example is that the surfaces will be raised by about 20°C and that the oil temperature will be raised by more than 100°C. These figures are in reasonable agreement with CROOK's[11,13] experimental estimates.

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5. CONCLUSIONS

The first aim of this paper has been to present a condensed version of the flash temperature theory in a form suitable for the experimentalist. Two alternative equations, the high- and low-speed formulae, can be used to calculate the flash temperature. Which of these two equations is appropriate for a given set of experimental conditions is determined by the magnitude of the dimensionless parameter L. The treatment of this paper emphasises that L is a measure of the depth of penetration of the heat below the surface. The high- and low-speed equations then represent two extreme limiting conditions: at low speeds the heat flows through the whole of the constriction region, and at high speeds the penetration of heat is limited to such small depths that the flow of heat is linear.

Flash temperature theory uses an idealized model to represent a rubbing contact and it is important to enquire whether it is a reasonable approximation to practical conditions or whether some changes in the model are required. For example, one requires to estimate the effect of protective oxide films or the consequences of a sub-division of the contact. A rigorous solution of these problems would be very complex but using the simple approach of the present paper it is possible to deduce the main effects of the appropriate modifications to the theoretical model. For example, it is shown that under most practical conditions the presence of a protective surface film of low thermal conductivity raises the surface temperature but leaves the substrate temperature practically unchanged.

The graphical methods which have been described can greatly reduce the time and labour involved in obtaining an estimate of the flash temperature. They have additional advantages in the intermediate speed range ($0.1 < L < 5$) where neither the low-speed nor the high-speed equations can be applied with sufficient accuracy. The maximum advantage is obtained from the use of graphical methods, when, in a series of experiments, a single combination of materials is used and it is required to calculate the maximum possible flash temperatures for various loads and speeds. Normally it is necessary first to calculate (by obtaining the value of L for each load and speed) whether the low- or high-speed equations apply and then to calculate the temperatures from the appropriate formulae. In contrast, by using a graph (such as those of Fig. 3) the temperature is obtained directly from the given values of the load, speed and coefficient of friction.

The experiments which have been described have been selected to illustrate the use of flash temperature theory in the interpretation of wear experiments. A feature of the experiments with Perspex is that, even at speeds as low as 10 cm/sec, the effects of frictional heating are confined to relatively thin surface layers. In experiments with metals[14] this requires speeds of the order of 10^4 cm/sec. Thus, high thermal gradients are established in metals only at the highest speeds of sliding but with materials of lower thermal conductivity they are achieved at speeds normally encountered in laboratory experiments or engineering practice. Such thermal gradients probably occur in surface protective oxide films and their existence in elasto-hydrodynamic oil films has been discussed in Section 4 above. In the example of an oil film 1 μ thick it

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has been shown that the temperature difference within the film is easily the largest transient temperature in the contact region and it may be more than five times greater than the surface flash temperature.

ACKNOWLEDGEMENTS

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SPECIFICATION OF THE VISCOSITY OF LUBRICANTS

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SUMMARY

The correct choice of a motor oil, as regards viscosity, can be made theoretically on hydrodynamic considerations.

The theoretical procedure involves laborious operations necessary for the development of the Fourier series which gives the variation of the attitude angle.

By a more approximate procedure one arrives at a general relation:

$$\mu_1 = 4.7 \cdot 10^{-4} \text{ kg . sec . m}^{-2}, \text{ say}$$

$$5 \text{ cSt at } \sim 90^\circ\text{C},$$

which should be correct for all new car engines.

This value must be corrected for wear, which can also be done by calculus or graphically.

Finally a convenient procedure is given for expressing viscosity changes at high temperatures.

RÉSUMÉ

Le choix correcte des huiles à moteurs, en ce qui concerne la viscosité, peut être fait théoriquement en partant de certaines considérations hydrodynamiques.

A cette fin, on doit accepter au préalable l'hypothèse raisonnable que le palier de bielle est l'organe de frottement le plus sensible à la viscosité de l'huile.

Du point de vue théorique, le procédé exige certaines opérations assez laborieuses nécessaires au développement de la série Fourier, qui donne la variation de l'angle de la ligne des centres avec la direction de la charge.

Par un procédé plus approximatif, on arrive à une relation générale

$$\mu_1 = 4.7 \cdot 10^{-4} \text{ kg . sec . m}^2, \text{ respectivement}$$

$$= 5 \text{ cP}$$

qui devrait être valable pour tous les moteurs neufs d'automobile.

Cette valeur doit être corrigée pour l'usure, ce qu'on peut faire soit par calcul, soit par représentation graphique.

Dans la dernière partie du travail, on propose une méthode convenable pour exprimer des conditions de viscosité pour température élevée.

INTRODUCTION

Although the piston rings and cylinders are the parts of an engine subject to most rapid wear by friction, there are good reasons for considering the bearings and particularly the crankshaft pin bearings to be much more sensitive to the oil viscosity.

Indeed, the crankshaft pin bearings are obliged to support greater loads (commonly reaching 100 kg/cm²) of variable intensity and from varying directions, while owing to the unsatisfactory heat dissipation they are subjected to temperatures some

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30°C greater than those to which other engine bearings are subjected. They are thus the first to wear and the first to seize. But since the lubricating conditions are hydrodynamic, a suitable lubricant can be selected by a rigorous analytical procedure.

At the same time, theoretical and practical evidence would suggest[1] that the oil viscosity imposed by this machine part will certainly satisfy the requirements of any other machine part of the same engine; consequently, the choice of the lubricant as regards viscosity becomes a routine task, but not a very easy one.

In the following a procedure for calculating the necessary viscosity is described. It is based on recent theories[2] and is in good agreement with other well-known and classical theories.

SYMBOLS

h_a	average height of asperities	$\Delta\theta^*$	variation of θ^*
δ_2	minimum film thickness	ζ	duty parameter
Δ	radial clearance	μ_1	absolute viscosity at entrance
F	force	ψ	relative clearance
r	journal radius	C_n, C_t	notations for complex functions of α and λ
b	breadth of bearing		
λ	breadth/diameter ratio	R	crank throw
α	relative eccentricity	l	length of connecting rod
α_{nom}	nominal value of α	Ω_1	angular velocity
α_{max}	maximum value of α	Ω_3	pulsation of engine's cycle
θ^*	attitude angle	F_{co}, F_{sn}, F_{cn}	coefficients of the Fourier series
θ^*_{nom}	nominal value of θ^*		
θ_{min}	minimum value of θ^*		

CALCULATIONS

The first and the most important condition for hydrodynamic lubrication is that the minimum oil film thickness (δ_2) should be greater than the sum of the tallest asperities (h_0). Conventionally, for fine workshop practice the relation is considered to be:

$$\delta_2 \geqslant 3\, h_a \tag{1}$$

The relative eccentricity α is given by

$$\alpha = 1 - \frac{\delta_2}{\Delta} \tag{2}$$

where Δ is the radial clearance.

The attitude angle θ^* is readily obtained for large values of α, using the relation

$$\alpha = \cos\theta^* \tag{3}$$

When this is not possible, it would be necessary theoretically to use some rather difficult formulae. It is more convenient, however, to proceed graphically with the aid of diagrams deduced by the laborious procedure, like that of Fig. 1, which is theoretically correct for 0.5 breadth diameter ratio (λ).

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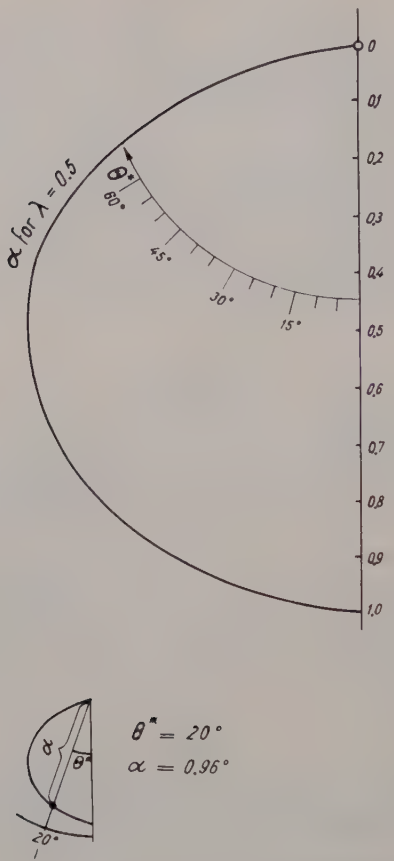

Fig. 1. Variation of relative eccentricity (α) against attitude angle (θ^*) for $\lambda = 0.5$.

The variations $\Delta\theta^*$ of the attitude angle θ^* result from the convenient Fourier series:

$$\Delta\theta^* = \frac{1}{2}\cdot\frac{R}{L}(\sin\Omega_1 t - 1) - \frac{\Omega_1}{\Omega_3}\cdot\frac{1}{2F_{co}}\sum\frac{1}{n}\left[F_{sn}\left(\cos n\Omega_3 t - \cos n\frac{\pi}{2}\cdot\frac{\Omega_3}{\Omega_1}\right) - \right.$$

$$\left. - F_{cn}\left(\sin n\,\Omega_3 t - \sin n\frac{\pi}{2}\cdot\frac{\Omega_3}{\Omega_1}\right)\right] \qquad (4)$$

where the coefficients F_{co}, F_{sn} and F_{cn} are obtained with the aid of the harmonic analyser from the diagram showing the magnitude of the resultant force on the particular engine crankpin during an entire cycle.

Thus we get

$$\theta^*_{cor} = \theta^*_{min} + \Delta\theta^* \qquad (5)$$

and further, using Fig. 1, we obtain the (corrected) value of α_{cor}.

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The duty parameter ζ results from

$$\zeta = \frac{3\sqrt{C_n^2 + C_t^2}}{1 + \alpha_{nom}} \tag{6}$$

where C_n and C_t are complicated functions of α and λ.

The graphical method again proves more convenient, as is seen in Fig. 2, corresponding to Fig. 1, which gives the calculated diagram of ζ against α for $\lambda = 0.5$.

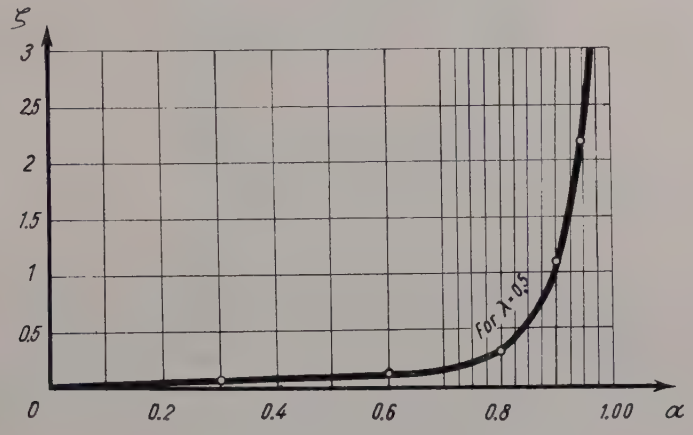

Fig. 2. Variation of duty parameter (ζ) against relative eccentricity (α) for $\lambda = 0.5$.

The absolute viscosity is related to the geometric and functionally known parameters as follows

$$\mu_1 = \frac{F \cdot \psi^2}{2br\Omega_1\zeta} \quad \text{kg . sec . m}^{-2} \tag{7}$$

where μ_1 is the absolute viscosity of the oil when entering in the bearing, and Ω_1 the representative speed of the friction assembly.

Objections can easily be raised to such a time-consuming method. To exemplify this, it is sufficient to mention the 25 symbols used and the supplementary 20–60 values of F_{co}, F_{sn} and F_{cn}. As will be shown, a great number of cumbersome operations can be omitted if suitable generalizations are made.

A practical generalizing approach

The height of asperities is an elusive and not very significant parameter. It has been shown[3] that after some running time the surface quality changes to a value corresponding to the particular mechanical conditions of every rubbing part, independent of the original surface quality. We are thus obliged to abandon eqn. (1). Since it has been proved that the hydrodynamic film can be very much thinner, if we assume surface qualities corresponding to the normal fine workshop practice, we can accept a safe general solution

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$$\delta_2 \simeq 1\,\mu \tag{8}$$

For Δ we can adopt the well-known, empirically verified solution

$$\Delta = 0.001r \ldots 0.0015\,r \tag{9}$$

Further, using the values $\delta_2 \simeq 1\,\mu$, $\Delta = 0.001r \ldots 0.0015r$ and eqn. (2), we readily obtain the value of α. We can also by-pass the mentioned value of Δ and use instead the values resulting from the workshop tolerances, which, for normal motor-car engines, would give for the crankshaft pin bearing:

$$\Delta = 0.03 \ldots 0.09 \text{ mm} \tag{10}$$

It follows that

$$\alpha = 1 - \frac{\delta_2}{\Delta} = 0.97 \ldots 0.99 \text{ mm} \tag{11}$$

The corresponding attitude angle is given by Fig. 1

$$\theta^* \simeq 7^\circ \tag{12}$$

The difficult task of finding $\Delta\theta^*$ was solved by applying eqn. (4) for a number of representative engines. In fact we used a slightly modified procedure described in another paper[1]. In Figs. 3a, 4a and 5a are given the resultant force on crankpin diagrams for three engines investigated, namely M-20, ZIS-120 and Horch 3 A, which served for calculating the Fourier coefficients of eqn. (4). The corresponding

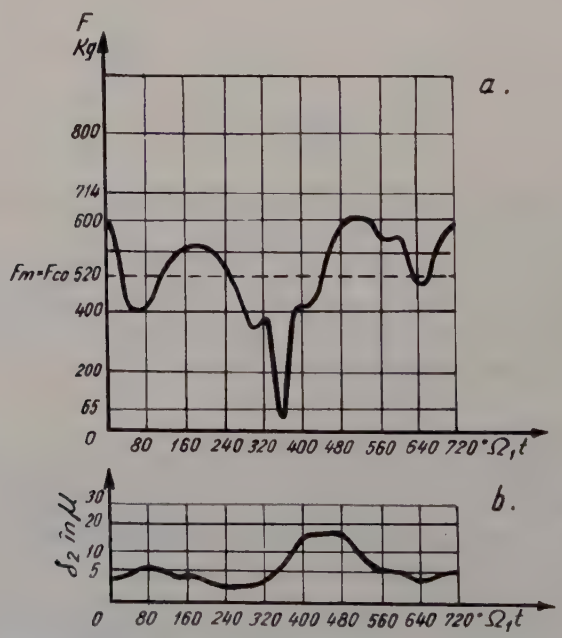

Fig. 3. Variation of resultant force (F) on crank pin against crank angle (θ^*) and the corresponding oil-film thickness (δ_2) in microns, for engine M-20.

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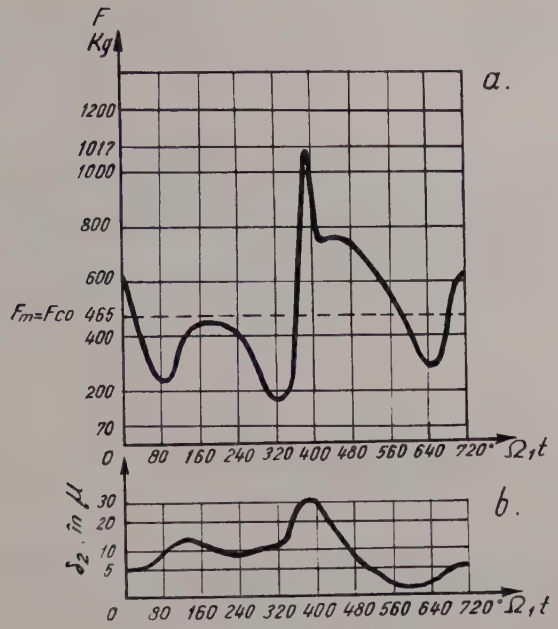

Fig. 4. Variation of resultant force (F) on crank pin against crank angle (θ^*) and the corresponding oil-film thickness (δ_2) in microns, for engine ZIS-120.

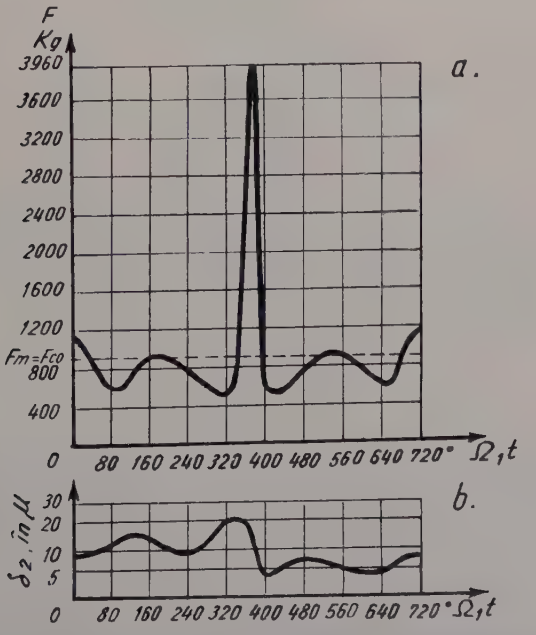

Fig. 5. Variation of resultant force (F) on crank pin against crank angle (θ^*) and the corresponding oil-film thickness (δ_2) microns, for engine Horch 3 A.

values of $\Delta\theta^*$ introduced in eqn. (5) furnished, with the aid of eqn. (3), the values of α necessary for calculating δ_2 during a complete cycle (Figs. 3b, 4b and 5b)

$$\delta_2 = \Delta\,(1 - \alpha)$$

where for Δ we adopted the average value of condition (9), *i.e.* $\Delta = 1.25r \times 10^{-3}$. The calculated values of $\Delta\theta^*$ were 30°, 23.35° and 20° respectively. For other engines, the values were intermediate. Since there are no practical objections, we can adopt the most unfavourable condition

$$\Delta\theta^* = 30° \tag{13}$$

and

$$\theta^*_{cor} = \theta^* + \Delta\theta^* \simeq 37° \tag{14}$$

Again using Fig. 1, we find the corrected value of α

$$\alpha_{cor} \simeq 0.875 \tag{15}$$

The duty parameter ζ results from Fig. 2, which can be applied to crankshaft pin bearings, which usually have $\lambda = 0.5$. We obtain

$$\zeta = 0.9 \tag{16}$$

The problem is now reduced to the solution of the previously mentioned relation for μ_1 (eqn. 7), which we shall write for convenience:

$$\mu_1 = \frac{F}{2br\Omega} \cdot \psi^2 \cdot \frac{1}{\zeta} \tag{17}$$

For the investigated engines we found that

$$\frac{F}{2br\Omega} \simeq 1400 \tag{18}$$

an average value which can be somehow justified since $F/2br$ is the pressure on the bearing, limited by the mechanical strength of the bearing material, and $F/2br\Omega$ corresponds to a pressure parameter corrected for speed.

The relative clearance ψ by definition and convention is

$$\psi = \frac{\Delta}{r} = \frac{0.001r}{r} \ldots \frac{0.0015r}{r} = 0.001 \ldots 0.0015 \tag{19}$$

The value of ψ must be corrected for the thermal dilatation of the bearing, because the working temperature is appreciably higher, but varying in a rather small range around 90°C. Following the procedure indicated by another author[4], we obtain the corrected value of ψ

$$\psi_{cor} \simeq 5.5 \cdot 10^{-4} \tag{20}$$

The value of ζ was already stated (eqn. 16), so that we can write the final form:

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$$\mu_1 \simeq 1400\,(5.5 \cdot 10^{-4})^2 \cdot \frac{1}{0.9} \simeq 4.7 \cdot 10^{-4}\,\mathrm{kg.sec.m^{-2}} \qquad (21)$$

or

$$\mu_1 \sim 5\ \mathrm{cP\ at} \sim 90^\circ\mathrm{C}$$

Correcting for wear

The dimensions used for deducing this viscosity are those of a new engine, but an engine is new only for a short time. Wear begins with running-in and continues as shown in Fig. 6. The nearly linear part AB is particularly interesting. It has the significance of a constant wear rate and corresponds in practice to the useful life of the machinery.

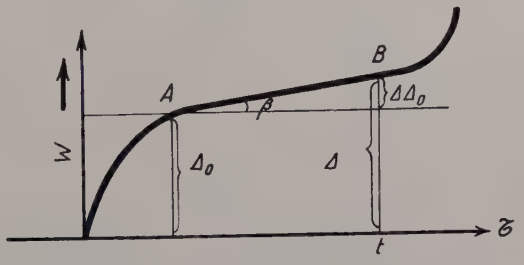

Fig. 6. Conventional variation of wear (w) against time (τ).
Δ_0 — original radial clearance
Δ — radial clearance at time t
$\Delta\Delta_0$ — increase of original radial clearance
A — wear after running-in
B — wear at time t

The AB portion can be easily expressed:

$$\Delta = \Delta_0 + \tau \tan\beta \qquad (22)$$

where τ is time of wear and $\tan\beta$ is the slope of the wear line.

Dividing by r we get

$$\frac{\Delta}{r} = \frac{\Delta_0}{r} + \frac{\tau \tan\beta}{r} = \psi = \psi_0 + \frac{\tau \tan\beta}{r} \qquad (23)$$

Further we can write

$$\tan\beta = \frac{\Delta\,\Delta_0}{\tau} \qquad (24)$$

so that eqn. (24) becomes

$$\psi = \frac{\Delta\,\Delta_0}{r} + \psi_0 \qquad (25)$$

The value of $\Delta\Delta_0$ can be deduced statistically and represents the maximum wear admitted for that machinery before overhauling. For instance, for car engines, it is believed that the average bearing wear before overhauling is about 0.025 mm[5].

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The time taken to reach that wear limit differs according to local conditions, but for a certain set of conditions this time should be reproducible.

For an arbitrary moment t, the wear will be

$$\text{wear} = \frac{1}{r} \cdot \frac{0.025t}{\tau} \tag{26}$$

where τ is the time limit for the maximum permissible wear.

Considering $\tau = 1$, the value of t/τ will vary between 0 and 1.

Further we can use the fraction $1/r$ to simplify the graphical construction, where it has the significance of the wear line's slope, as is shown in Fig. 7, because

$$\frac{1}{r} = \frac{\psi_0}{\Delta_0} \tag{27}$$

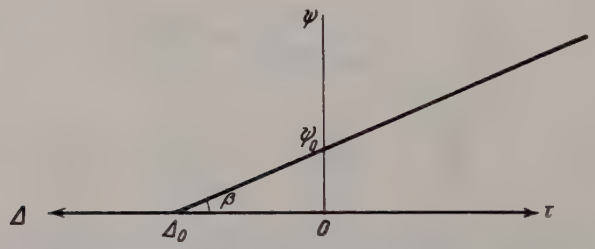

Fig. 7. Increase of relative clearance (a) against time of wear (τ) as a function of original radial clearance (Δ_0). $\tan \beta$ = slope of increase line.

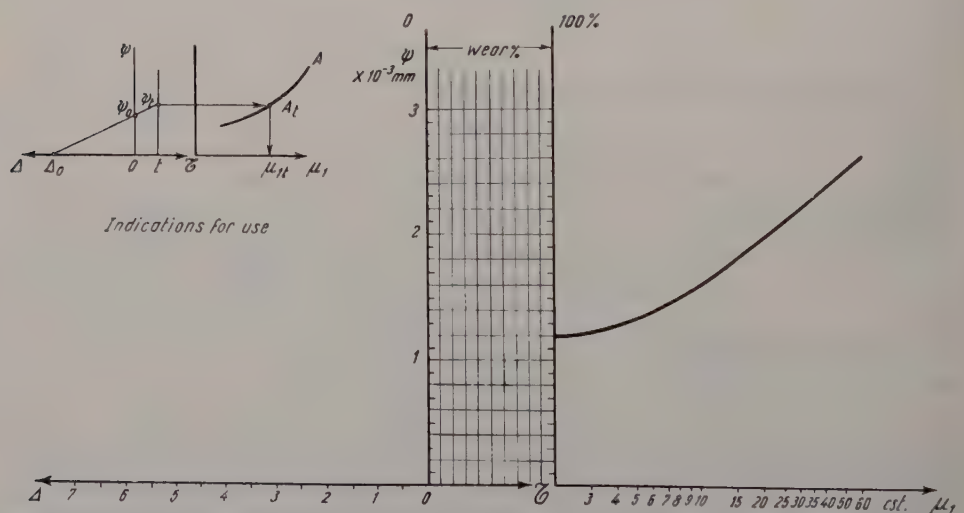

Fig. 8. Nomograph for deducing the lubricant viscosity (μ_1), function of orgiinal radial clearance (Δ_0), original relative clearance (ψ_0), and wear (%τ).

Now we can deduce any value of ψ, at any moment, and we can calculate the corresponding value of μ, for car engines, using relation (21) in its new form:

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$$\mu_1 \doteq 1550\ \psi_t^2\ \text{kg.sec.m}^{-2} = 1.53 \cdot 10^8\ \psi_t^2\ \text{cP} \tag{28}$$

The mathematical development can be conveniently replaced by the nomograph of Fig. 8. The values of ψ_t (relative clearance at time t) are given by the intersection between the $\Delta_0\psi_0$ line and the normal to the $o\tau$ axis, passing through point t. From the point ψ_t we trace a horizontal line up to the point A_t. The intersection of the normal from this point to the μ_1 axis gives a point μ, which is the desired viscosity in cP.

Easier still and very useful as a rough guide is Table I, compiled for normal car engines.

TABLE I

Finish	*Desired viscosity of oil at working temperature (in cP)*		
	Initial wear	*Normal wear*	*Worn-out*
Good		5	10
Fair	5	10	17.5
Poor	10	17.5	35

Influence of high temperatures

The present method for defining analytically the viscosity at 100°C (210°F) is of no use if we wish to know the viscosity at other temperatures of up to 20–30°C higher or lower. For this reason, we considered it more reasonable to express the viscous behaviour of the oil at high temperature by another parameter related to a known variation of the lubricant viscosity.

For this purpose, we found it convenient[1] to use the viscosity (ν_c) and the temperature (t_c) at which

$$\left|\frac{d\nu}{dt} \cdot \frac{1}{\nu}\right| = 0.02 \tag{29}$$

or, in other words, which corresponds to a viscosity variation of 2%/°C.

The value of T_c (the value of t_c expressed in absolute Kelvin degrees) can be readily deduced from the classical Walter Ubbelohde relation

$$\log \log (\nu + c) = a \log T + b \tag{30}$$

which served, as was shown[7], to obtain the equation

$$\log T_c = \frac{\log \dfrac{x}{a \ln 10} - b}{a - 1} \tag{31}$$

where

T_c = the absolute temperature corresponding to the adopted value of x, °K

x = the adopted value of the viscosity variation of the oil, eventually 0.02 as in eqn. (29)

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a and b = the Walter Ubbelohde constants of eqn. (30) for that particular oil
ln 10 = 2.3026.

If we know T_c, we pass over easily to the corresponding ν_c using eqn. (30) or the ASTM Chart.

The ν_c and t_c parameters present some interesting properties which have been discussed elsewhere[7]; we shall merely recall here the outstanding fact that it dispenses us of the a, b and c constants of eqn. (30), and as we can make the approximation

$$-\frac{d\nu}{dt}\cdot\frac{1}{\nu} = e^{(ax+b)} + c \tag{32}$$

by integration we find

$$\nu = \nu_c \cdot e^{-0.006(t-t_c)} \cdot e^{1.006\left[e^{-0.01604(t-t_c)} - 1\right]} \tag{33}$$

where

ν = the desired viscosity, cS
ν_c = the viscosity at which $|d\nu/dt \,.\, 1/\nu| = 0.02$
t = the temperature corresponding to ν, °C
t_c = the temperature corresponding to ν_c, °C

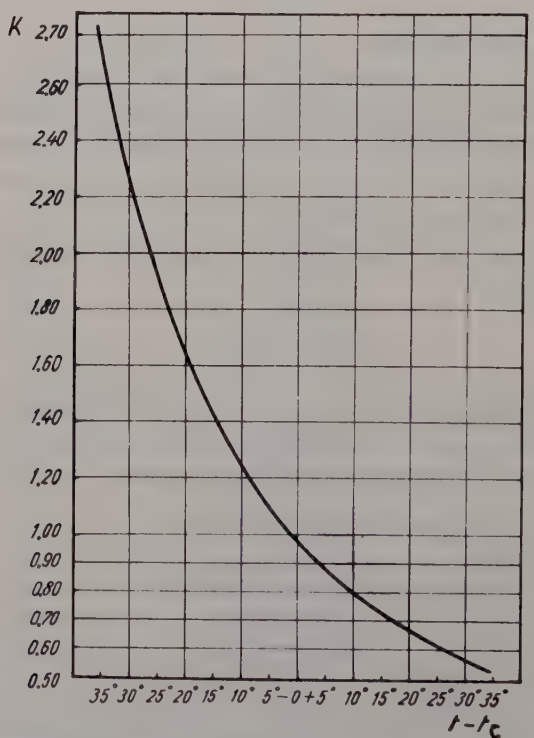

Fig. 9. Variation of k against temperature difference $(t - t_c)$, for calculating the viscosity at any temperature t (ν), if we know the viscosity (ν_c) and temperature (t_c) for which

$$\left|\frac{d\nu}{dt}\cdot\frac{1}{\nu}\right| = 0.02; \quad k = e^{-0.006(t-t_c)} \cdot e^{1.006\left[e^{0.01604(t-t_c)} - 1\right]}$$

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As the interesting temperature range is limited at some 20° or 30°C above or below t_c, the temperature-dependent term can be calculated and tabulated so that if we know ν_c and t_c, we can deduce the viscosity at any temperature, just by multiplying ν_c by a factor k corresponding to the desired temperature difference; this difference is considered negative if t is lower and positive if t is higher than t_c. The value of k is given in Fig. 9.

To simplify matters, we give also in Table II the values of ν_c and t_c for some common engine lubricating oils.

TABLE II

For oils with V.I. 100				*For oils with V.I. 50*			
Viscosity at 210°F (98.9°C) (cS)	ν_c *(cS)*	t_c *(°C)*	*SAE rating*	*Viscosity at 210°F (98.9°C) (cS)*	ν_c *(cS)*	t_c *(°C)*	*SAE rating*
6	6.8	93.4	10	6	5.6	101.85	10
7	6.9	99.5		7	5.8	106.85	
8	7.1	104.5	20	8	6.0	110.95	20
9	7.5	106.8		9	6.17	114.25	
10	7.8	109.8	30	10	6.38	117.15	30
12	8.4	113.9		12	6.74	121.70	
14	8.9	117.7	40	14	7.10	125.40	40
16	9.4	120.7		16	7.33	128.85	
18	9.8	123.7	50	18	7.57	131.80	50
20	10.2	126.2		20	7.77	134.35	

The values are only approximately correct, but satisfactory for practical purposes.

CONCLUSIONS

Engine lubricants can be selected, as regards viscosity, by an analytical procedure. A simpler method is proposed, based on the analytical one, in which reasonable generalizations were operated.

In practice, the oil viscosity should vary, at the working temperature, from 5 cP for good-quality new engines to 35 cP for poor-quality worn-out engines.

We make no mention of other oil functions, involving the problem of additives, widely discussed in the literature, on which we have done no original work.

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THE CONTACT BETWEEN A LOCOMOTIVE DRIVING WHEEL AND THE RAIL*

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SUMMARY

In connection with improving the adhesion of locomotives, a study was made of the conditions of contact existing between locomotive driving wheels and the rail. Observations were made of such measurable quantities as contact area and shape, relative movement or "creep" in rolling, and limiting coefficient of friction, with different values of vertical load, wheel diameter, tractive force, etc. The results obtained were compared with the calculations of HERTZ, CARTER, and others. Whenever possible measurements were made upon actual wheels and rails, but when this was impossible the problem was simulated by a model in a material giving greater deflection under load. Consideration was also given to the effect of wear upon tyres and rails. In general, reasonable agreement was obtained between actual and calculated values, except that the areas of contact could apparently be increased by roughness of the contacting surfaces, and in the case of creep there appeared to be some additional factor which has so far been neglected in calculation.

ZUSAMMENFASSUNG

KONTAKTE ZWISCHEN TRIEBRAD EINER LOKOMOTIVE UND SCHIENE

Um eine bessere Adhäsion von Lokomotiven zu erreichen wurden verschiedene Bedingungen für Kontakt zwischen Triebrädern und Schiene untersucht. Eine Reihe messbarer Grössen wurden bestimmt, nämlich Form und Umfang der Kontaktfläche, relative Bewegung oder "Kriechen" beim Rollen, sowie die Grenzwerte des Reibungskoeffizienten bei verschiedenen Normaldrucken, Raddurchmessern, Zugkräften usw. Die Ergebnisse wurden mit Berechnungen von HERTZ, CARTER u.A. verglichen. Wenn möglich wurden die Messungen an wirklichen Rädern und Schienen ausgeführt, andernfalls wurde das Problem an einem Modell nachgeahmt und zwar mit Material das unter Druckbelastung eine grössere Formänderung besass. Auch der Einfluss des Verschleisses auf "Reifen" und Schienen wurde berücksichtigt. Im allgemeinen stimmten gemessene und berechnete Werte redlich überein, jedoch kann anscheinend die Kontaktfläche durch Rauhigkeit der berührenden Oberflächen vergrössert werden, ausserdem scheint beim "Kriechen" noch ein weiterer Faktor aufzutreten, der bei den bisherigen Berechnungen nicht berücksichtigt wurde.

The original purpose of railways was to transport goods from place to place with the minimum expenditure of labour and money, but nowadays the intensity of traffic requires loads of all kinds to be conveyed as rapidly as possible, and with high rates of acceleration and deceleration. Modern designs of locomotive have therefore high power-weight ratios, with corresponding facilities for braking, so that the traffic offering can be carried without undue delay over the lines available. For this reason it is often found that electrification is of great value for increasing

* (Based on an opening address given to the Conference on Contact Stresses convened by the Institute of Physics in London, 1958.)

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the capacity of crowded traffic routes owing to the greater power that can temporarily be provided by a given weight of equipment.

The performance of any locomotive, whether steam, diesel, or electric can now be accurately forecast under any particular conditions of working, but its whole output, represented by the tractive effort exerted at the rim of the driving wheels, must be transmitted to the train through what is, in effect, a form of friction drive often working under very unfavourable conditions. Any failure of adhesion between the driving wheels and the rails will represent at best a reduction in the haulage capacity of the locomotive, or under severe conditions may even result in a failure of operation. With the greater power now available from modern electric locomotives this problem is becoming increasingly acute, and is consequently regarded as of major importance amongst the investigations at present being carried out by British Railways in connection with electrification.

While this problem is being studied from a number of aspects, one natural line of approach has been a study of the contact between a locomotive driving wheel and the rail, in an attempt to understand the mechanism by which the tractive force is transmitted. Actual contact between a wheel and the rail occurs only over a very small area, usually measuring only a fraction of a square inch. These contacts not only support the weight of the locomotive, usually some 10 tons per contact, but also transmit the tractive force which may amount to some $2\frac{1}{2}$ tons per contact. Since slipping is more likely to occur under wet conditions it follows that, even at these high pressures, some water must remain between the contact surfaces. While the surface of the wheel tyre is curved in the longitudinal direction, that of the rail is curved in the transverse direction. The contact may therefore be regarded as that between two surfaces relatively curved in each direction with or without a fine layer of water between them.

In view of the high pressures involved and the difficulty of access it was impossible to study the problem directly by many of the means that would ordinarily be considered. The approach has therefore been to study such external characteristics as could reasonably be measured, and to relate the conclusions to the corresponding results of theoretical analysis.

The theoretical basis for the calculation of the conditions of contact between two curved bodies was originally published by HERTZ[1]. HERTZ showed that the area of contact between such bodies pressed together was dependent upon the elastic properties of the materials and their effective curvature at the point of contact. In the case of an unworn tyre resting upon a new length of rail (Fig. 1), the undistorted surfaces may be regarded as those of two cylinders at right angles, the radius of the wheel R_1 providing all the curvature in the longitudinal direction, while the radius of the head of the rail R_2 represents the total curvature in the transverse direction. Provided the rail is in contact with the tapered portion of the tyre the transverse radius of the tyre R'_1 is infinite, and the longitudinal radius of the rail is, of course, also infinite. The fact that the surface of the tyre is actually coned is compensated by the corresponding inclination of the rail, and it is clear from geometrical

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considerations that the nature of the contact is independent of the lateral position of the wheel relative to the rail.

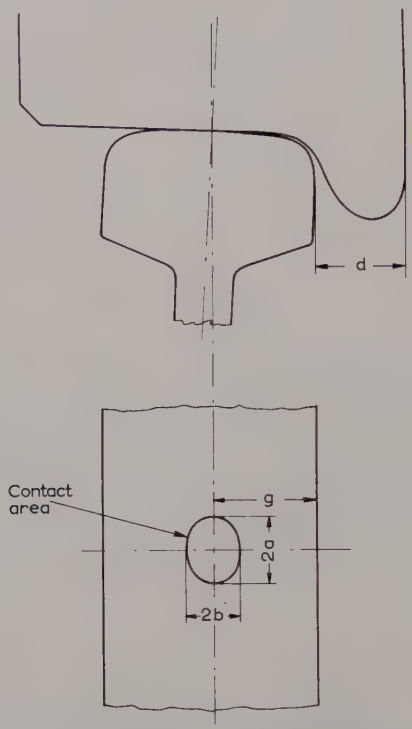

Fig. 1. Diagram of contact area between wheel and rail (showing nomenclature).

The essential features of the Hertzian method of calculation as applied to this problem are as follows: – Let the curvature of the wheel be represented by $2A$, and that of the head of the rail by $2B$. The relationship between these two curvatures which are at right angles may be represented by an auxiliary angle θ whose value is defined by

$$\cos\theta = \frac{B - A}{A + B}$$

The area of contact between these two bodies will be an ellipse, as shown in Fig. 1, the semi-axes of which are denoted by a longitudinally and b laterally, and these are calculated as follows

$$a = m \sqrt[3]{\frac{3\pi}{4} \cdot P \cdot \frac{k_1 + k_2}{A + B}}$$

$$b = n \sqrt[3]{\frac{3\pi}{4} \cdot P \cdot \frac{k_1 + k_2}{A + B}}$$

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in which P represents the perpendicular load between the two bodies, and k_1 and k_2 are constants representing properties of the materials of the tyre and rail respectively, being determined by

$$k = \frac{1 - \sigma^2}{\pi E}$$

where E is the modulus of elasticity and σ is Poisson's ratio for the material concerned. m and n are constants depending only on the angle θ, but as these functions are rather difficult to calculate they have been tabulated by HERTZ[1]. The area of contact is, of course, the area of the ellipse whose semi-axes are respectively a and b, which is πab.

It is thus seen that the dimensions of the area of contact should be proportional to the cube root of the applied load. The area of contact is also a function of the radius of the wheel, the effective radius of the head of the rail, and the materials of which the rail and tyre are composed. These last may be effectively represented by assuming that the value of the modulus of elasticity E for the steel of both the rail and the tyre is 13,400 tons per sq.in., and that Poisson's ratio σ for both materials is 0.3, so that the constants k_1 and k_2 have both the value

$$k = \frac{1 - \sigma^2}{\pi E} = \frac{1 - 0.3^2}{\pi \times 13{,}400} = 2.16 \cdot 10^{-5} \frac{\text{in.}^2}{\text{tons}}$$

HERTZ has also shown that, for the conditions described above, the distribution of pressure over the area of contact may be represented as a semi-ellipsoid constructed over this area. The maximum intensity of pressure is therefore at the centre, and has a value $1\frac{1}{2}$ times the average value of pressure. It is interesting to observe that, owing to the increase of area with load, the intensity of pressure by no means increases in proportion to the load.

As it seemed to be too difficult to make direct measurements of the intensity of pressure over the area of contact, it was decided to attempt first the measurement of the area of contact itself. The values obtained could then be compared with corresponding values calculated from the loads and dimensions involved, so that, if reasonable agreement could be obtained over a range of loads and wheel diameters, the validity of the Hertzian method of calculation would be established in this particular case. Moreover, since in this method, the calculation of the pressure distribution is intimately associated with that of the area of contact, success in relating actual and calculated values of contact area would afford reasonable justification for assuming the validity of the same calculation for determining the distribution of pressure. Agreement on these lines has been frequently obtained in the case of ball and roller bearings, where calculations of this sort are regularly employed, but in other cases it has been found[2] that, owing to the nature of the materials involved, the actual conditions of contact may differ from those calculated by the method of Hertz, though the general relationships of the quantities involved remained unimpaired.

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It was first necessary to establish a reliable method of measuring the contact area between the tyre and the rail. This proved unexpectedly difficult and a number of alternative methods had to be rejected before success was obtained. The method ultimately adopted was the very simple one of raising the tyre from the rail and inserting thin sheets of carbon and ordinary paper between the points of contact. On lowering the tyre, the intense pressure compressed the paper and caused the carbon to be transferred to the ordinary paper over the area of contact so that, on removing the tyre, an accurate imprint of the contact was obtained on the carbon paper which could subsequently be measured. The obvious error introduced by the thickness of the papers themselves was examined by making measurements of one particular contact with papers of different thickness and relating the measurements obtained to the thickness of the papers. In this way it was established that the fine papers employed introduced only a small percentage of error. An advantage of this method of reproducing the contact areas was that the impressions obtained could be further reproduced for permanent record by direct photographic means.

The first stage of the investigation was to obtain short lengths of new rail and tyre, to locate them appropriately in a hydraulic press, as indicated in Fig. 1, and to apply various values of vertical load, the area of contact being recorded for each value of load. These records were subsequently measured by planimeter, and the values of area so obtained are plotted in relation to the vertical load in Fig. 2.

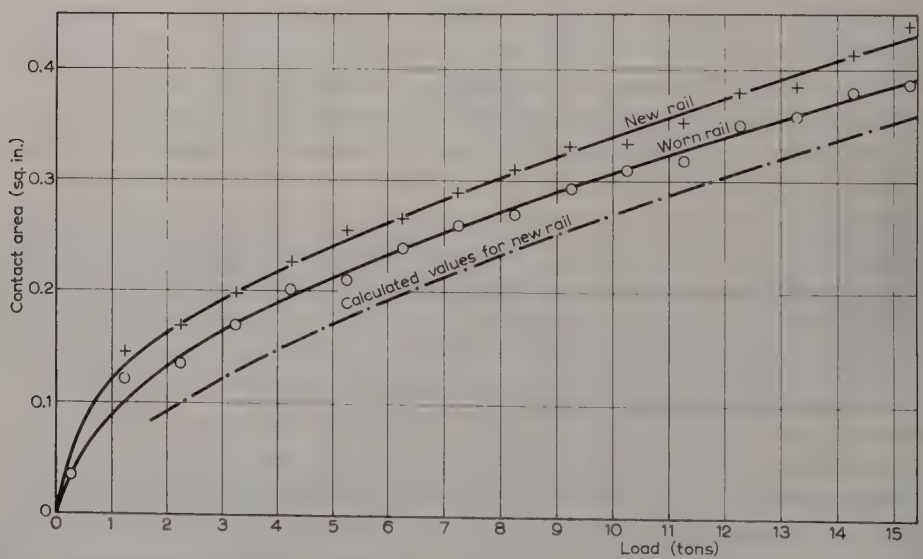

Fig. 2. Measurements in hydraulic press. Conical tyre in contact with new and worn rail. Relationship between contact area and load. Wheel diameter 49.75 inches.

Unfortunately in service neither tyres nor rail can maintain their profiles exactly, and, as shown by DEARDEN[3], the original radius of 12 in. at the head of a new rail is gradually reduced to 10 in. by a process of wear. The new rail was therefore replaced

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by a short length of worn rail taken from service and the experiment repeated, the corresponding observations also being plotted in Fig. 2.

These measurements of contact area are compared in Fig. 2 with corresponding values calculated as described above, and it is seen at once that, while the theoretical relationship is clearly maintained, the actual values of area are greater than those calculated, a conclusion which is in line with the conclusions of BERNDT[2]. Examination of the detailed dimensions revealed that this increase was shared more or less proportionately between the two principal axes in the case of the new rail, whereas that with the worn rail, even allowing for the change in curvature, was almost entirely due to increase in the major axis.

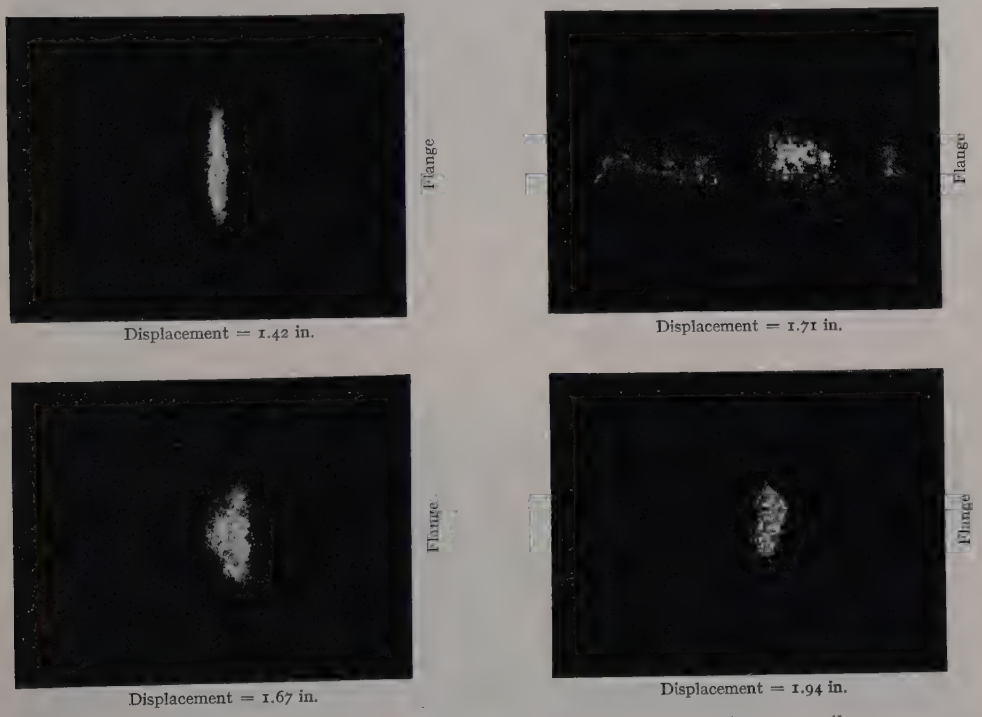

Fig. 3. Wheel–rail contact areas between worn tyre and worn rail.

The case of worn tyre was somewhat more complicated because any departure of the tyre surface from the conical makes the area of contact dependent upon the lateral position of the tyre relative to the rail. A wheel having a worn tyre was therefore placed in contact with a length of worn rail under a fixed value of load and a number of impressions were taken with the tyre in different lateral positions as indicated by the dimension d in Fig. 1. Typical impressions so obtained for various values of d are shown in Fig. 3, in which it is seen that a narrow impression obtained in one extreme position gradually broadens until a wide but somewhat uneven

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contact is obtained near the mid-position, but narrows again as the other extreme position is approached. As the lateral position of a wheel relative to the rail is rarely stationary in running, it follows that the area of contact and the intensity of pressure are probably continuously varying while the train is in motion.

It was next desired to repeat as much as possible of the preceding observations under actual working conditions, and this was done under the tyres of a wheel and axle assembly taken from a motor coach of the Liverpool St.–Shenfield electric stock, which was placed on a length of track at the Ilford Car Shed. Different values of load were obtained by taking impressions under the tyres of the wheels and axle alone, as part of the bogie, with the motors in position, and with the body resting on the bogies, the vertical force at the tread of the wheels being determined in each case from the known weights of the vehicle. Although some difficulty was caused by turning marks on the tyres, the measurements of area obtained, shown plotted against the load in Fig. 4, gave better agreement with the values calculated by the method of Hertz.

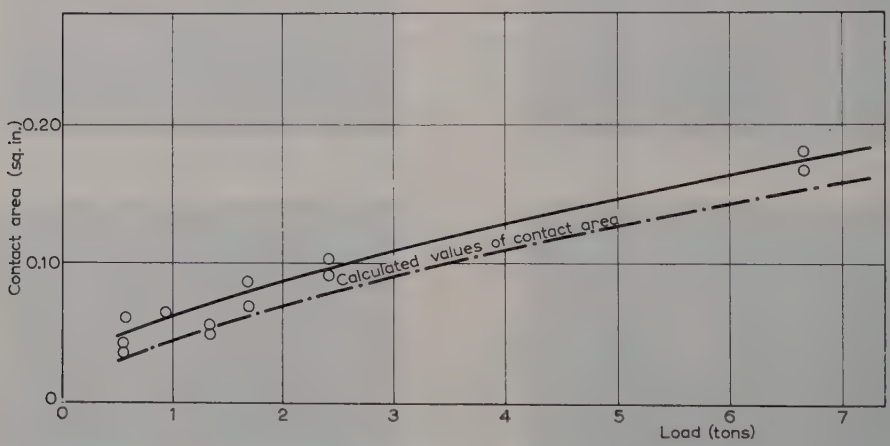

Fig. 4. Measurements under wheel and axle assembly with different loads. Turned tyre in contact with new rail. Relationship between contact area and load (wheel diameter 42 in.).

The next stage was to study the relationship between the area of contact and as wide as possible a range of wheel diameters. Normally, however, the range of wheel diameters carrying any appreciable load upon a railway is somewhat limited, so very large and very small wheels were sought to provide the required information. Fortunately a pair of nominally 8 ft. 6 in. wheels, believed now to be the largest existing, were available in the retired single-driver locomotive "Cornwall" (L.N.W.R. No. 3020) seen in Fig. 5, while a pair of 2 ft.6 in. wheels were to be found as trailing wheels of certain of the 0–4–2 tank locomotives, seen in Fig. 6, which are employed as shunting locomotives in Crewe works. This part of the investigation was therefore carried out at Crewe, wheels of intermediate sizes being provided by locomotives

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Fig. 5. Locomotive "Cornwall" (L. & N.W. R. no. 3020). Driving wheels 8 ft. 6 in. diameter, the largest wheels available for this investigation.

Fig. 6. Crew Works shunting locomotive. Trailing wheels 2 ft. 6 in. diameter, the smallest wheels available for this investigation.

coming out of the works, some having freshly turned and others slightly worn tyres.

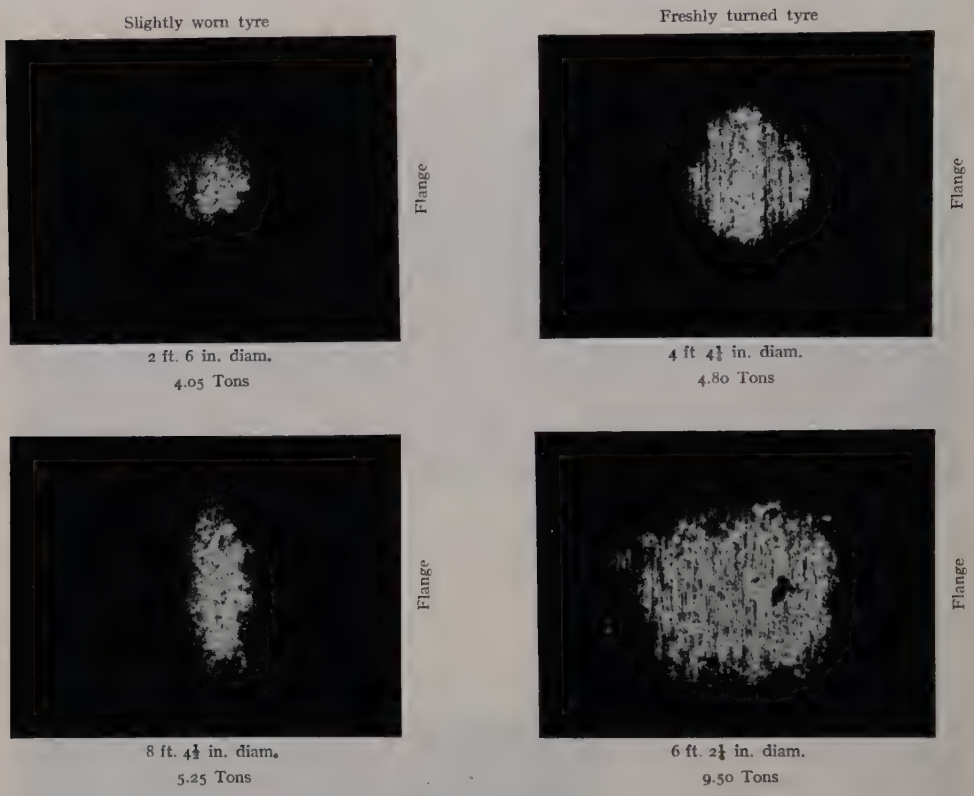

Fig. 7. Wheel–rail contact areas.

Typical impressions of contact area obtained in this manner are shown in Fig. 7, in which it is clear that the marking of the freshly turned tyres has a broadening effect upon the contact area, presumably with corresponding reduction in average intensity of pressure. The areas of contact are therefore better compared by the measurement of the longitudinal semi-axis a, and these values, corrected to a value of wheel load of 7 tons, are shown in Fig. 8. The general relationships between the dimensions measured and the wheel diameter are again in agreement with that calculated by the method of Hertz, though, as before, the values are somewhat higher, an effect that is further increased when the tyres are freshly turned. A corresponding plotting of values of the transverse semi-axis b showed that the broadening effect, and consequently the increase in area, increases rapidly with increase in diameter of the wheel.

The conclusions drawn from these observations were therefore that with new tyre and new rail the measured areas of contact generally agreed with, but were somewhat

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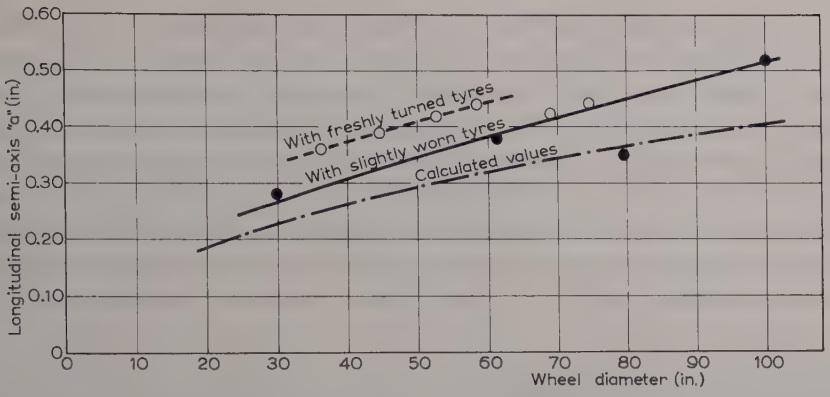

Fig. 8. Measurements with variable wheel diameter. Slightly worn or freshly turned tyre in contact with new rail. Relationship between longitudinal semi-axis a (corrected for 7 tons wheel load) and wheel diameter.

greater than the corresponding values calculated by the method of Hertz. It may therefore be presumed that the distribution of pressure over this area is also as forecast by Hertz, but of slightly less value. Wear of the rail increases the intensity of pressure as the contact area is reduced due to the increased curvature of the rail surface, whereas wear of the tyre may reduce the pressure and increase the area of contact, but this is dependent upon the lateral position of the tyre relative to the rail. The parallel between the increase of contact area produced by the markings of newly turned tyres and the increase in area over the theoretical value which has been consistently noted throughout these observations suggests that the latter is at least partly due to irregularities in the contacting surfaces of the same nature as, but much smaller than, the turning marks. When the wheel rolls on the rail the record of the area of contact is extended, as shown in Fig. 9, which corresponds with the tread markings observed on the heads of rails in service.

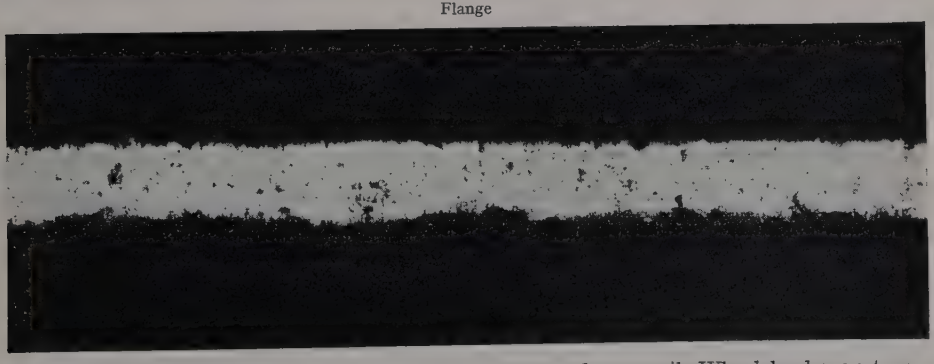

Fig. 9. Contact surface of rolling wheel. Worn tyre and new rail. Wheel load 1.35 tons.

The next step was to consider the contact after the application of tractive effort to the wheel. This problem has been studied analytically by MINDLIN[4], PORITSKY[5],

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and JOHNSON[6], who give expressions for determining the stresses at the contacting surfaces. Each of these found that the contact area was slightly displaced in the direction of the tractive force, but without appreciable change of shape. In practice, however, it was found to be extremely difficult to obtain experimental corroboration of these analyses since with normal railway materials the displacements produced were very small in relation to the other dimensions. Measurement of the contact area was no longer practicable by the methods described above, while the area involved was too great for measurement by the electrical methods used in the case of small electrical contacts.

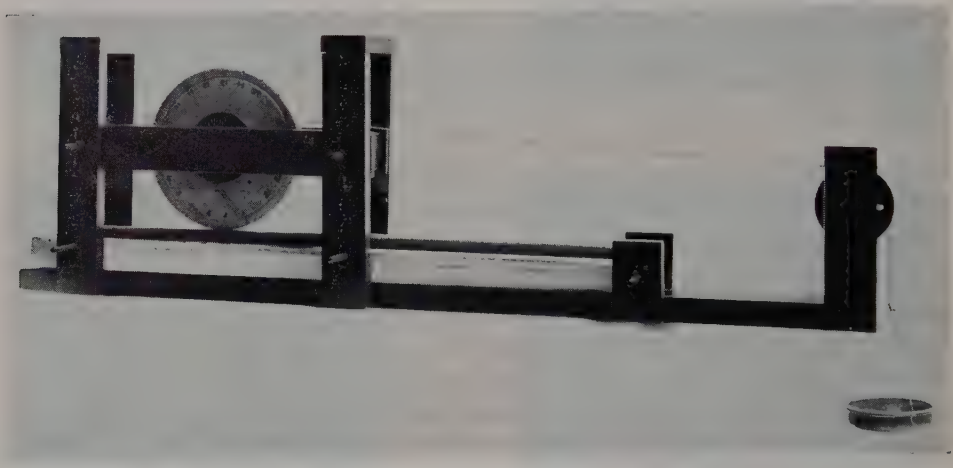

Fig. 10. Wheel–rail model.

As it did not seem possible to attack this problem directly, it was decided instead to construct a model in some elastic material in which comparatively large deflections could be produced by moderate values of force, and to study its behaviour under different values of load and tractive effort. This model, seen in Fig. 10, simulated the problem in the form of plane stress, the tyre of the wheel and the head of the rail being represented by shaped pieces of expanded polyurethane rubber having a very low modulus of elasticity. On test this material was found to give linear displacement in response to stress within reasonable limits. The wheel could be caused to roll along the rail by moving the lower carriage, and any desired value of load could be produced by adjusting the height of the carriage containing the "axle" bearings, the relation between height and load being determined in a preliminary test. Tractive effort could be applied to the wheel in either direction by placing weights in the scale pan, the string from which was wound in the appropriate direction round a drum immediately behind the wheel. The wheel was of 6.88 in. diameter, both wheel and rail having a width of 0.48 in., and the side surfaces were graduated for measurement.

The wheel was first applied to the stationary rail with different values of load and the corresponding lengths of the contact surface were measured. These lengths of

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contact were plotted against the values of the applied load and compared with corresponding values calculated from the properties of the material, when it was found that the relationships were similar but the actual values exceeded the theoretical values in much the same ratio as with the actual tyres and rail. The experiment was then repeated with different values of tractive effort applied to the wheel, but only negligible change of contact length was produced.

This latter experiment was then repeated with the difference that, after each application of load and tractive effort, the wheel was rolled a short distance along the rail and the observations made at some other position. In these circumstances the length of the contact surface was found to be reduced by a small amount roughly proportional to the tractive effort, reaching some 4% of the total length immediately before slipping at a moderately high value of load. The coefficient of adhesion at which slipping occurred, which was determined separately, appeared to decrease with increase of load, varying from over unity when the surfaces were pressed lightly together to about 0.8 at comparatively high loads.

When tractive effort is applied to a loaded wheel rolling along a rail, the deformation of the materials in the neighbourhood of the point of contact results in the wheel rotating a greater amount than would correspond geometrically with the distance travelled. This is known as creep, and is usually defined as the ratio of the difference between the two motions and the distance travelled. Creep is a function of both load and tractive effort, and has been investigated mathematically by CARTER[7], who reached the conclusion that the value of creep for similar elastic materials could be expressed in the form

$$\frac{\mu}{2R}\sqrt{\frac{8P(1-\sigma)R}{\pi G}}\left(1-\sqrt{1-\frac{T}{\pi P}}\right)$$

where T is the tractive force at the rim of the wheel with the radius R, P the vertical load, μ the limiting value of the coefficient of friction, σ is Poisson's ratio and G the modulus of rigidity of the materials. This expression has been later confirmed by PORITSKY[5].

CARTER also pointed out that the material of the tyre is stretched, and that of the rail compressed, immediately before contact, and claimed that after contact the two surfaces are locked together until the shear force between them is sufficient to overcome the friction, after which slipping occurs. Thus the area of contact comprises a forward area over which the two surfaces are rigidly held together, and a rearward area over which slipping is constantly taking place while the wheel is in motion.

According to the above expression there should be no creep when no tractive effort is applied, so measurements were next made on the model to discover whether any creep occurred during free rolling of the loaded wheel. The wheel was moved down to represent various values of load, and then rolled a given distance along the rail in each direction in turn, the relative positions of the marking on the tyre and rail being noted at the end of each movement. It was found that there was a definite

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rate of creep, in each case in the opposite direction to the motion, and this was related to the value of the load. Since there was some vertical deflection of the tyre at the point of contact it was at first supposed that this creep might be due to reduction of effective wheel radius, and values of creep were calculated in this way for different values of load. Fig. 11 shows the measurements obtained in relation to the load, together with the calculated values, and it is seen that, while there was an appreciable amount of creep without application of tractive effort, which may be called "geometric" creep, this was not as great as would have been expected from the deflection of the wheel rim at the contact.

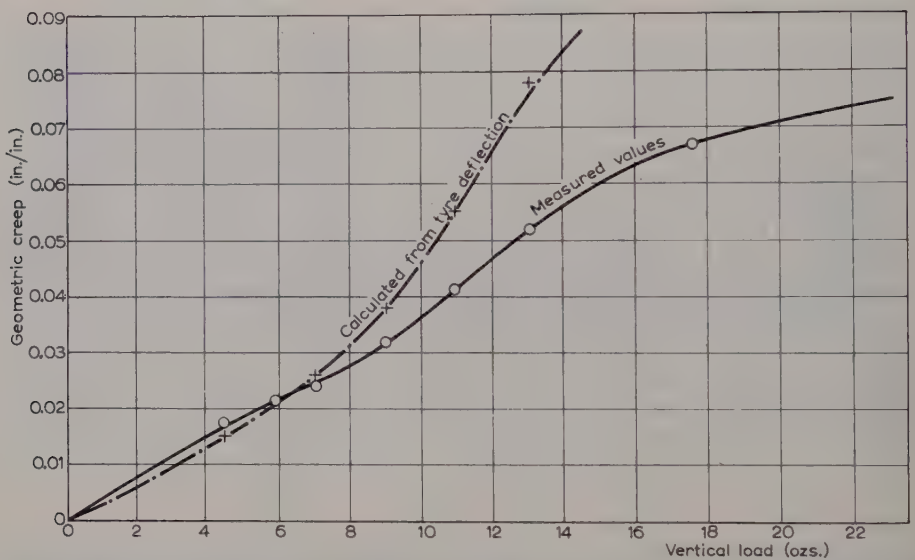

Fig. 11. Creep measured at rim of wheel when running freely (geometric creep). Creep is in each case in the opposite direction to motion.

The preceding experiment was then repeated with the application of various values of tractive effort for each value of load. As it appeared that the creep due to tractive force, or "torque" creep, was additional to the "geometric" creep, the values measured were first corrected for the "geometric" value of creep corresponding to the load, when it was found that the remaining "torque" creep was a function of both load and tractive effort. This "torque" creep was always in the direction of the tractive force at the rim of the wheel, but had alternative values depending upon whether the tractive force was in the positive or negative direction in relation to the movement, that is, whether the wheel was "driving" or "braking". The values of creep so determined with the tractive force in the positive direction are shown in relation to tractive effort in Fig. 12, and in relation to load in Fig. 13. Typical values of creep, measured at various values of vertical load, with the tractive effort in the positive and negative directions, are compared in Fig. 14.

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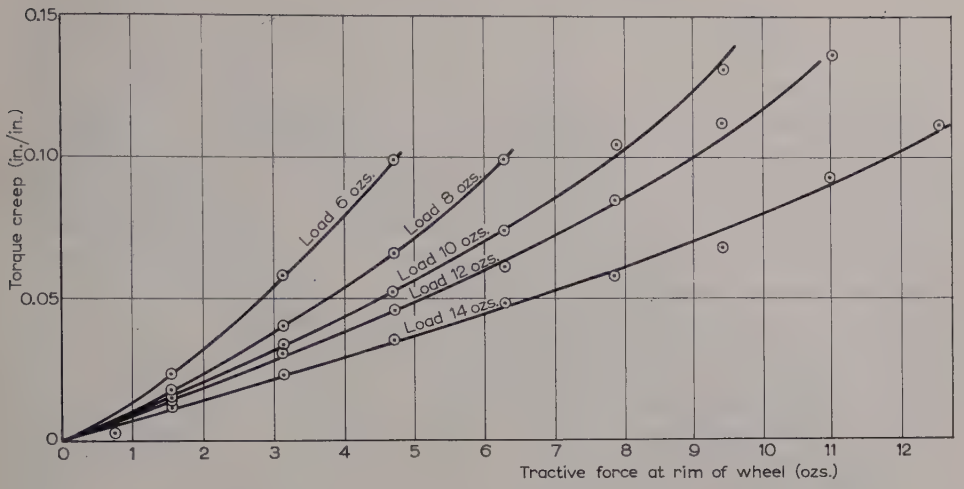

Fig. 12. Creep at rim of wheel due to tractive effort for different values of vertical load (additional to "geometric" creep). Creep is in each case in direction of tractive effort at rim of wheel.

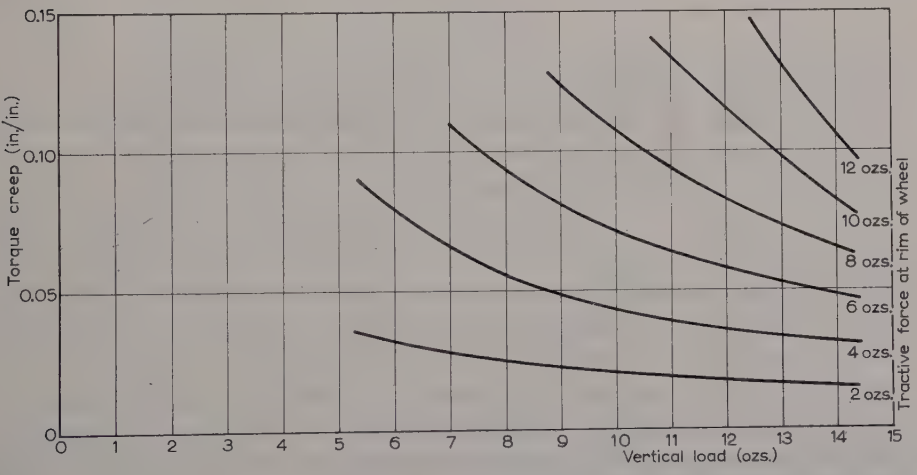

Fig. 13. Creep at rim of wheel due to tractive effort. Relationship between creep and load for different values of tractive force (additional to "geometric" creep). Creep is in each case in direction of tractive effort at rim of wheel.

The nature of the expanded rubber employed in these experiments was such that it was impossible to make direct comparison with actual conditions involving a steel tyre and rail, but the indications obtained from the model are so definite that they may be regarded as pointers for subsequent investigation under actual conditions, or, in certain cases, may offer explanations of facts already observed. The existence of "geometric creep" was unexpected, not having been forecast theoretically. It is, of course, possible that this factor may also be a property of the material employed,

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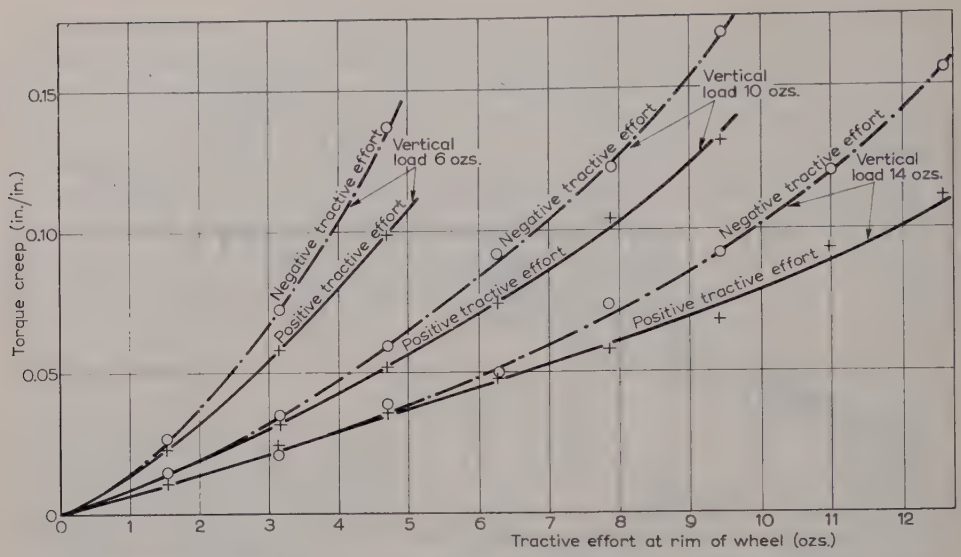

Fig. 14. Comparison of creep at rim of wheel due to positive and negative tractive effort (additional to "geometric" creep). Creep is in each case in direction of tractive effort at rim of wheel.

and may be due to hysteresis. Nevertheless it is now desirable that such a factor should be looked for when observations are made under actual conditions.

The results obtained, Figs. 12, 13 and 14, are of the form forecast by CARTER, but are not in numerical agreement with his method of calculation. An interesting mathematical suggestion in this connection has been made by Mr. C. STOREY. In any given conditions of working there exists, according to CARTER, a direct relationship between tractive force and creep. Thus for any value of "geometric" creep there may be postulated a hypothetical value of tractive force which would produce a corresponding result. If then, instead of adding the values of "geometric" and "torque" creep, their corresponding values of tractive force are added and the value of creep recalculated, much closer agreement can be obtained between the calculated and actual values. At present it is difficult to understand the significance of this hypothetical tractive force, but probably the real inference to be drawn is that, while there is obviously agreement in principle between the work of CARTER and the actual observations, there must be some factor which occurs in free rolling which has been omitted from the calculation.

Finally it was decided to measure the limiting coefficient of adhesion of locomotive driving wheels in running to see whether there was any variation of adhesion with speed. This was done on one driving axle of an electric locomotive, the circuit of which was rearranged so that the motors of the other axles were disconnected, and tractive force could be gradually applied to one axle only. Slipping of the test axle was indicated by a differential slip indicator showing the difference in movement between the driven and one of the undriven axles. This locomotive was hauled by another locomotive, and was attached to the British Railways Mobile Testing Plant

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by means of which constant speed could be maintained independently of the tractive force produced.

By running the train at various values of constant speed and gradually increasing the tractive force applied to the test axle until slipping of the wheels occurred, the limiting value of tractive force could be observed for each value of speed. The values so obtained were rather spread and were obviously dependent upon the amount of moisture present on the upper surface of the rails. The values for completely dry and completely wet rail were therefore segregated and average values obtained for each speed, as shown in Fig. 15. It is seen from these results that there is a definite reduction in the limiting value of tractive force at the wheel contact with speed, and that, while the limiting values of force are greatly affected by the presence of moisture at the contact, the rate of reduction with speed is but little affected. It is, of course, probable that the contact is influenced by vibration or transient changes of load that will be increasingly experienced with increase of speed.

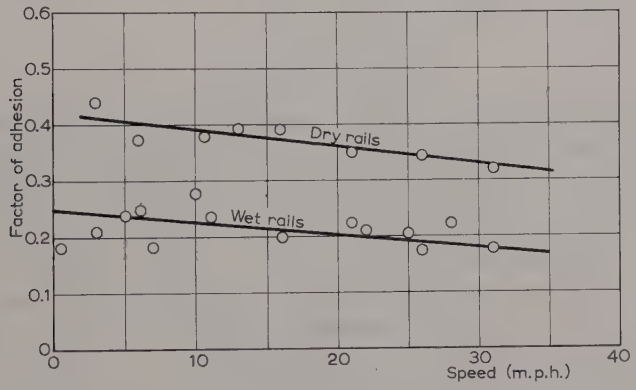

Fig. 15. Average value of adhesion on wet and dry rails measured on one axle of locomotive no. 26034 at constant speed.

While the observations described represent only an introduction to the study of wheel–rail contacts, there are certain conclusions which may be drawn that will be of assistance in further investigation.

In the first place the area and dimensions of the contact between new or slightly worn wheel and rail are roughly in accordance with the calculations of HERTZ except that, for some reason, the area is usually a little larger than expected and the stress correspondingly reduced. The distribution of load over the contact area is presumably ellipsoidal as represented by HERTZ.

The nature of the contact at the wheel tread is seriously affected by slight roughness of the contacting surfaces, the areas of contact being almost doubled in the case of freshly turned tyre. Since perfection of finish cannot be expected in either tyre or rail surfaces this probably at least partially explains the increase in contact area normally observed.

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The impression of a wheel upon a rail produces such deflection of the materials that some measure of creep occurs even when no tractive force is applied to the wheel. When tractive force is applied there is additional creep produced at the wheel tread in accordance with the theory of CARTER, but it seems that some factor associated with free rolling has been neglected in his calculation. The attention of mathematicians is drawn to this conclusion in the hope that a fuller analysis of this problem may be forthcoming.

In the case of a worn tyre the area of contact, and consequently the distribution of load, is greatly affected by the lateral position of the wheel relative to the rail. This may well be a factor of consequence in the study of the riding of bogies.

Since the limiting coefficient of adhesion is so greatly reduced by the presence of water, it is clear that some amount of moisture must remain between the contact surfaces even under pressures of the order of 50 tons per sq. in. Understanding of the nature and action of whatever film of moisture is able to exist under these circumstances would undoubtedly be of great help in dealing with the problems of rail adhesion with which railways are now faced.

ACKNOWLEDGEMENTS

Acknowledgement is gratefully made to the British Transport Commission for permission to publish this article, to Mr. P. LARKAM who carried out many of the experiments, and to Mr. C. STOREY for his help in the mathematical analysis of the results.

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Received February 10, 1959

Authors' Abstracts

Considerations on the Fundamentals of Wear Processes

L. ROZEANU AND O. PREOTESCU – *Studii și cercetări de mecanică aplicată*, 9 (1) (1958) pp. 169–185; (in Rumanian), (10 fig.; 10 ref.).

Every wear process implies an energy input and an energy balance of the type

$$\eta_u = \frac{1}{\tilde{a}\frac{A}{\sigma}\cdot\frac{Q}{V}+1}$$

where η_u is the energetic efficiency of the wear process, A is the mechanical equivalent of heat, σ is the free energy of newly formed surfaces, Q is the heat generated, V is the volume of worn material, and $\tilde{a}$ is a coefficient, with the dimension of length, having the significance of V/S, where S is the area of newly formed surfaces.

This expression states in a simple way the relation between the nature of the material (σ), the intensity of the wear process ($\tilde{a}$) and the nature of the wear process (Q/V).

As regards the nature of the wear process, retaining – for the moment – the three fundamental forms of wear (contact, mechanical, chemical) for usual intensities, we can arrive at the following relative values: contact wear $Q/V \sim 10^8$, chemical wear: $Q/V \sim 10^7$, mechanical wear $Q/V \sim 10^6$.

Every type of wear is influenced in a particular way by the local conditions and this influence is clearly shown by the variation of $\tilde{a}$. Sometimes, there is a change from one type of wear to another and then Q/V is greatly modified.

This new way of expressing the wear is particularly useful for evaluating the oiliness of a system and for comparing the relative merits of various testing machines and testing methods.

A Qualitative Separation of Wear Factors

L. ROZEANU AND O. PREOTESCU – *Rev. mécan. appl.*, 2 (1) (1957) 115–127; (in English), (10 fig.; 12 ref.).

The paper contains a qualitative, general analysis of wear mechanisms and wear factors. The following conclusions are drawn.

By over-simplification it is possible to separate four fundamental types of wear, due respectively to adhesion, abrasion, corrosion and fatigue. Adhesive wear, although practically inevitable, seems to be far less dangerous under normal running conditions than is predicted by theory. Abrasive wear must be considered a severe accidental type of wear which can be partially controlled and almost completely eliminated. Corrosive wear is a most acute form of wear, but by careful treatment the corrosion may prove to be an important auxiliary in the fight against wear. Fatigue wear is an inevitable and steadily growing type of wear. It cannot be eliminated but can be partially reduced. It is an undesirable form of wear under any conditions.

The time factor was used to define three chronological sets of conditions of wear: initial (running-in), normal (useful life) and final(run-out machines). The running-in seems to be a complement of production technology for correcting local faults under conditions which easily promote damage. No smoothing of the entire friction surface is likely to take place.

Normal conditions are those resulting from the complex action of statistically normal factors in the process of friction. Final conditions are those in which deleterious factors appear in a well developed state, after having exercised an invisible activity in the two foregoing phases of wear conditions.

Considerations about Wear of Friction Parts During Running-in

L. ROZEANU AND O. PREOTESCU – *Studii şi cercetări de mecanica aplicată, 8* (1) (1957) 85–109; (in Rumanian), (19 fig.; 8 ref.).

It is a common belief that running-in means smoothing of working surfaces.

The experimental findings are not in agreement with this belief and theoretically it is shown that by a 1/3 reduction of the centre-line average value of the asperities heights, the increase of the metal content in the oil would be hardly noticeable by current analytical procedures.

At the same time, we usually find dimensional losses where by weight no difference is noticed and vice-versa.

This shows that running-in means rather a technological process for correcting those faults that have escaped quality control or those that are imposed by actual production necessities.

The Standardization of Wear Tests

L. ROZEANU – *Standardizarea, 11* (1) (1959); (in Rumanian).

As is well known, wear is the visible result of a great number of factors acting by quite different mechanisms.

While the number of mechanisms is limited to four fundamental types, the number of intervening factors is very large indeed, and their intensity varies in a still larger range.

Nevertheless, for the intensities of any factor the following classification can be made:

— possible intensities due to the surroundings;
— probable intensities connected with the actual stage of the industrial development or the operating range;
— critical intensity which marks the beginning of catastrophic wear or of any wear at all.

When discussing wear or testing for wear, we must choose the conditions so that the amount of wear is significant. The study of critical intensities must precede any testing procedure.

Tool Wear and Machinability

E. M. TRENT (Hard Metal Tools Ltd., Coventry) – *J. Inst. Production Engrs., 38* 1959; (27 pp., 36 fig., 16 ref.).

The study of worn cemented carbide tools with the aid of the microscope has shown the main types of wear, and factors affecting wear, including flank wear, the built-up edge, cratering, deformation, mechanical chipping and thermal cracking. The first four of these wear factors have been studied separately and their occurrence over a wide range of conditions on a number of ferrous materials has been investigated by a series of short time cutting tests. The results have been presented in the form of machining charts, in which cutting speed and feed rate are the co-ordinates to logarithmic scales and the occurrence of build-up, cratering and deformation is plotted. The value of such charts in summarising and ordering a large amount of information on tool wear and machinability has been demonstrated. *An important aspect of machinability, the range of speed and feed over which normal flank or clearance face wear is the predominant form of wear and which could be called "the most useful cutting range", is particularly clearly shown on these charts.* These can supplement existing information on rates of flank wear.

The flank wear occurring in short time cutting tests has also been studied. Two factors having great influence on the rate of wear appeared to be chemical composition of the work material and tool material and the presence of a built-up edge on the tool.

As regards chemical composition of the work material, the element having the greatest effect was sulphur which was present in the form of manganese sulphide and greatly reduced the rate of flank wear. The elements chromium and nickel played a relatively small part unless nickel was present in sufficient quantities to give an austenitic structure, when the rate of wear increased greatly.

It is possible for the alloying elements introduced into steel to exercise a direct effect on the rate of tool wear by forming compounds which act as abrasives or as lubricants between the tool and the work material.

There is, however, another and probably more important way in which alloying elements can exert an indirect influence on tool wear and machinability.

Micro-examination of sections through the built-up edge has suggested the process by which this may protect the tool surfaces from wear. The complete absence of tool wear under some conditions suggests that, under these conditions the flow is entirely within the metal being cut, the layer of metal in contact with the tool surface being stationary. Under other conditions the metal in contact with the surface is stationary on part of the interface and in motion over the remainder. It is concluded from this that *the nominal surface velocity of the work material relative to the tool may bear no relation to the true velocity at the interface on the rake or flank surfaces.*

There is a complex pattern of flow around the cutting edge, changing with the cutting conditions. The types and rates of wear depend on the details of this flow pattern. It seems probable therefore, that elements such as sulphur and nickel exert their greatest effect by altering this flow pattern, nickel by forming austenite and sulphur by forming manganese sulphide acting largely as an "internal lubricant" in the steel.

Superimposed on the flow pattern and related to it is a pattern of steep thermal gradients. Until more is known of the complex patterns of flow and temperature in the work material near the cutting edge, it is unlikely that it will be possible to account for the observed rates of wear in terms of the properties of work material and tool and the measured cutting forces.

Evaluation of Dry Powdered Lubricants at 1000°F in a Modified Four-ball Wear Machine

S. L. COSGROVE, L. B. SIBLEY AND C. M. ALLEN (Battelle Memorial Institute, Columbus, Ohio) – *ASLE Preprint 59 AM 1A-1.*

The development of advanced aircraft systems has introduced new problems in lubrication. There has been some success with certain solid lubricating compounds in high-temperature sliding experiments and in bearings reported by PETERSON AND JOHNSON[1], GLAESER, ALLEN, AND GOLDTHWAITE[2], and others [3-10]. Thus, it seemed desirable to explore the possibilities of lubrication by solids at elevated temperatures.

The criteria used in the selection of wear-resistant materials for this study were hot hardness, oxidation resistance, and resistance to thermal and mechanical shock. It is felt that these properties are most important for high-temperature components such as bearings and gears.

The prospective solid lubrication compounds were evaluated at 1000°F for their over-all friction and wear behavior in a modified four-ball machine.

General performance of materials

Under the conditions of the initial series of lubricant evaluations in the four-ball apparatus, using the cobalt-base alloy, only two lubricants—lead oxide and metal-free phthalocyanine—exhibited relatively low wear (less than 1-mm scar diameters). The friction was moderate with lead oxide (coefficient af about 0.15), while metal-free phthalocyanine exhibited a low initial coefficient of friction (0.06) but a higher over-all friction (coefficients up to 0.3). Molybdenum disulfide lubricated the cobalt alloy with a low initial friction that lasted longer but the wear was much more drastic (2.5-mm scar diameter) than would be expected even though this experiment was operated over twice as long as the experiments with the other lubricants. Of course, the lubricants that exhibited poor friction and wear behavior with the cobalt-base alloy in the initial evaluations may have done better with some other wear-resistant materials. However, the preliminary research seemed to demonstrate that these three lubricants—metal-free phthalocyanine, lead oxide (litharge), and molybdenum disulfide—had enough potential lubricating ability to be chosen for further study.

Under conditions of lubricant replenishment and higher sliding speed, and with four wear-resistant materials, metal-free phthalocyanine showed lower initial coefficients of friction than either molybdenum disulfide or lead oxide. However, equilibrium friction coefficients were lower, in several other cases, notably with lead oxide as a lubricant. With AISI Type 440 C stainless steel, the iron-base alloy, and the cermet material, metal-free phthalocyanine gave the lowest wear. With the cermet, particularly small (0.62) wear scars were observed. With the tool steel both molybdenum

disulfide and lead oxide gave appreciably smaller wear scars than did metal-free phthalocyanine.

The results of this preliminary study indicate that satisfactory lubrication of rubbing surfaces at 1000°F may be obtained with low friction and wear by the appropriate application of certain solid lubricating compounds. However, further development of prospective lubricant pretreatment and replenishing techniques is required before specific lubrication systems can be recommended.

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The Causes and the Control of Electrical Currents in Bearings

J. BOYD AND H. N. KAUFMAN (Westinghouse Research Laboratories, Churchill Borough, Pittsburgh 35, Pa.) – *Lubrication Eng., 15* (1959) 28–35; (11 fig., 1 table, 11 ref.).

When bearings become damaged in service, the question often arises as to whether the damage was due to electrical current. The damage caused by electrical currents may be negligible or may be catastrophic. Such currents may occur from several sources: (1) A bearing may carry current as a necessary part of an electrical circuit; (2) the current may be self-induced as a result of the design characteristics of the machine, or (3) the current may result from an electrostatic phenomenon. This article describes the various sources of electrical current, the damage resulting from such currents, and the methods by which the damage may be reduced or eliminated.

Systematic Abstracts of Current Literature

Selected from Battelle Technical Review 1959 and from other sources

1. Deformation and Fracture

1.1. *Fracture*

Etat actuel du problème de la rupture fragile.
Present State of the Brittle Fracture Problem. G. M. Boyd. *Revue de la Soudure*, v. 14, no. 4, 1958, p. 175-187.
Study of one of the causes of the ruptures of mild steel structures, due to fatigue, or fatigue and corrosion. Characteristics of brittle fractures, inception, effects of temperature, relationship between strength and resistance, welding and protection.

A Review of Intergranular Fracture Processes in Creep.
P. W. Davies and J. P. Dennison. *Institute of Metals, Journal*, v. 87, pt. 4, Dec. 1958, p. 119-125.
Slow deformation of metals at elevated temperatures is known to lead to the development of intergranular cracks. Both qualitative and quantitative aspects of the theories relating to such internal fissuring are reviewed, and it is shown that different processes become operative according to the prevailing conditions of stress and temperature.

The Markings on Fracture Surfaces.
Eugene F. Poncelet. *Society of Glass Technology, Journal*, v. 42, Dec. 1958, p. 279T-288T + 8 plates.
Fracture surfaces on glasses exhibit characteristic marking which can be interpreted in terms of the author's theory of propagation of fracture; this theory is summarized briefly and is illustrated by a collection of photographs of various forms of fracture surfaces.

On the Strength of Classical Fibres and Fibre Bundles.
B. D. Coleman. *Journal of the Mechanics and Physics of Solids*, v. 7, no. 1, Nov. 1958, p. 60-70.
Discusses tensile strength distributions for fibres whose strength is independent of the rate of loading. Reasons are presented for expecting that the tensile strength should obey the Weibull distribution. The statistical theory of the strength of bundles of these fibres, as developed by Daniels, is applied to infinite bundles composed of fibres which obey the Weibull distribution. It is found that the ratio of the tensile strength (units of force at break per initial unit area) of a bundle to the mean tensile strength of the constituent filaments decreases monotonically with increasing dispersion in the strength of the constituent filaments.

1.2. *Fatigue*

Internal Stresses and Fatigue in Metals.*
A. M. Freudenthal. *Physics Today*, v. 12, Feb. 1959, p. 16-19.
A symposium on Internal Stresses and Fatigue in Metals was held Sept. 3-5, 1958, at the General Motors Research Laboratories. Invited papers were presented by research workers in solid-state physics, metal physics, and engineering from England, France, Germany, and the United States. The singular feature of the conference is the attempt to clarify some of the basic aspects of the two phenomena by considering them in the light of concepts of solid-state physics or physical metallurgy rather than in empirical engineering terms alone, and thus to promote a deeper understanding of the interaction of the phenomena by the "interaction" of their metal physics and engineering aspects. It is surprising to note how rapidly, through this interaction, the extensive progress in metal physics has transformed the engineer's approach to the properties of the materials he uses, and has provided him with the beginning of that "sense of structure" which Cyril Smith considers to be the characteristic mark of the modern approach to the mechanics of materials, as distinct from the conventional or "classical" approach based on the mechanics of the solid isotropic continuum.

Some Quantitative Information obtained from the Examination of Fatigue Fracture Surfaces.
D. A. Ryder. Royal Aircraft Establishment (Gt. Brit.) September 1958. 6 pp., diagrs., photos. (Ask for N-69106)
Examination of the fracture of a programmed fatigue specimen has demonstrated that the

* The Proceedings of this symposium, edited by G. M. Rassweiler and W. L. Grube, have been published by Elsevier Publishing Co. Amsterdam, 1959, 451 pp., 280 fig., 9 tables.

characteristic striations first observed by Zapffe and Worden are each associated with a single cycle of stress. Quantitative microscopy on the fractures of a series of Rolls Royce fatigue specimens showed the mean spacing of the striations to be increased by increase of fatigue stress. The second part of the paper deals with the examination of a fracture from a programmed fatigue test on a large bolted joint. From measurements of the distance between progression markings, the crack propagation rates in two fatigue areas were plotted. The results demonstrated that cracks were propagating in this specimen very early in its fatigue life. It is thought that the early propagation of cracks in this specimen was due to the severe fretting damage that occurred during the test.

Some Corrosion Fatigue Observations on High Purity Aluminum-Zinc-Magnesium Alloy and Commercial DTD 683.
C. A. Stubbington and P. J. E. Forsyth. Royal Aircraft Establishment (Gt. Brit.) May 1958. 15 pp., diagrs., photo's., tabs. (Ask for N-66304)
Rolls Royce rotating cantilever air and corrosion fatigue tests have been made on the above materials which have been aged for various times. These results are compared with special reference to the form of fracture and microstructure. A mechanism for corrosion fatigue in these alloys is discussed.

Effect of Lubricant Viscosity and Type on Ball Fatigue Life.
William J. Anderson and Thomas L. Carter. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 266-272.
Four paraffinic mineral oils with viscosities of 5 to 113 centistokes at the 100°F test temperature were used. Longer life was obtained with more viscous oil, life varying approximately as the 0.2 power of lubricant viscosity. In investigation of methyl silicone, a paraffinic mineral oil, a sebacate, a water base glycol, and an adipate were used. Each had a viscosity of about 10 centistokes at the 100°F test temperature. The 10% life was about 40 times as great with the silicone as with the adipate.

Fatigue of a Nut and Bolt.
P. B. Walker. Royal Aircraft Establishment (Gt. Brit.) June 1958. 22 pp., diagrs. RAE Structures 238. (Ask for N-66130)
The factors that adversely affect the fatigue life of a tension bolt are analysed. The aim is to produce a comprehensive picture, derived as far as possible from fundamental principles and showing the various factors in relation to each other. Both external loading and the internal stress pattern are considered. The effects of plastic deformation and of controlled pretensioning are also brought into the general picture as complicating factors, though usually beneficial.

Improve Fatigue Strength With Abrasive Tumbling.
Iron Age, v. 183, no. 8, Feb. 19, 1959, p. 129-131.
Sponsored by the Abrasive Grain Association, the test program proves the usefulness of tumbling in attaining better fatigue strength.

2. Friction

2.1. *Miscellaneous*

The Viscosity Wedge.
A. Cameron. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 248-253.
Equations are set up and solved numerically giving coefficients for oil film thickness and friction. Thickness is comparable (three quarters) of that carried by Michell pads. Computed results are applied to a parallel surface thrust bearing described by Fogg and shown to give tolerable agreement with experimental findings.

Plane Sliders of Finite Width.
Donald F. Hays. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 233-240.
Illustrates the manner by which the separation of variables in Reynolds' equation may be effected, completes the solution of the resulting eigenvalue problem, and shows the formulation of the expressions for load capacity, frictional resistance, center of pressure, coefficient of friction, and rates of flow. Typical pressure hills are shown as well as approximate flow patterns throughout the slider area.

2.2. *Tire-to-surface*

Reibung von Gummi.
Friction of Rubber.
Eric Gough. *Kautschuk und Gummi*, v. 11, Nov. 1958, p. WT303-WT312.
Reasons for using rubber in structural elements and power transmission. Frictional properties. Skidding at low speeds. The use of belts and disks. Transmission of traction and brake pressure through tires. Cornering. Friction laws. Comparison between rubber and metal friction.

Tire-to-Surface Friction especially under Wet Conditions.
Richard H. Sawyer, Sidney A. Batterson, and Eziaslav N. Harrin. *NASA Memorandum 2-*

23-59L, March 1959. 13 pp., diagrs., photos.
Results of the tire-to-surface friction measurements in braking on runways under various surface conditions are presented and compared for a C-123B airplane and a friction cart. Comparisons of wet-surface friction studies with a 12-inch-diameter tire, an automobile tire, and an extra-high-pressure aircraft tire are also given.

Low Tire Friction and Cornering Forces on a Wet Surface.
Eziaslav N. Harrin. *NACA TN 4406*, September 1958. 27 pp., diagrs., photos.
An investigation was made by the National Advisory Committee for Aeronautics to study typical tire behavior on wet runways. A specially constructed tire treadmill served as a tire test vehicle whereas force measurements were made by means of a straingage balance. Values of maximum braking, full-skid braking, and rolling friction coefficients were obtained at yaw angles of 0° and 4° while operating two tire types (smooth and diamond treaded) in water depths from 0.02 to 0.09 inch and at speeds up to 94 feet per second including tire planing. Cornering-force coefficients were measured at the free-roll and maximum braking conditions.

Study of Taxiing Problems Associated with Runway Roughness.
Benjamin Milwitzky. *NASA Memorandum 2-21-59L*, March 1959. 12 pp., diagrs.
This paper briefly summarizes available statistical data on airplane taxi operations, examines the profiles and power spectra of four selected runways and taxiways covering a wide range of surface roughness, considers (on the basis of theoretical and experimental results) the loads resulting from taxiing on such runways over a range of speeds and, by synthesis of the aforementioned results, proposes new criteria for runway and taxiway smoothness which are applicable to new construction and may also be used as a guide for determining when repairs are necessary.

Investigation of the Maximum Spin-up Coefficients of Friction obtained during Tests of a Landing Gear having a Static-Load Rating of 20,000 Pounds.
Sidney A. Batterson. *NASA Memorandum 12-20-58L*, January 1959. 24 pp. diagrs., photos., tab.
An experimental investigation was made at the Langley landing loads track to obtain data on the maximum wheel spin-up coefficients of friction developed during landing. The forward speeds ranged from 0 to approximately 180 feet per second and the sinking speeds, from 2.7 to 9.4 feet per second.
The results indicated the variation of the maximum spin-up coefficient of friction with forward speed and vertical load.

2.3. *Skin friction*

Measurements of Skin Friction in a Plane Turbulent Wall Jet.
A. Sigalla. *Royal Aeronautical Society, Journal*, v. 62, Dec. 1958, p. 873-877.
Considers the inner layer of fluid between the wall and the position where the velocity is a maximum as a boundary layer with an outer uniform free stream of velocity equal to the local maximum velocity.

Preliminary Investigations of Spiked Bodies at Hypersonic Speeds.
Seymour M. Bogdonoff and Irwin E. Vas. *Journal of the Aero/Space Sciences*, v. 26, Feb. 1959, pp., 65-74.
An examination of the flow over blunt bodies equipped with a spike to minimize heat-transfer rates. The use of a spike protruding from a hemispherical-nosed cylinder at M = 14 decreased the pressure level by an order of magnitude and the heat transfer to a fraction of that measured on a hemisphere without a spike. The general technique appears to hold considerable promise for hypersonic flight.

Use of a Stanton Tube for Skin-Friction Measurements.
S. S. Abarbanel, R. J. Hakkinen, and L. Trilling. *NASA Memorandum 2-17-59 W*, March 1959. 48 pp., diagrs., photos.
An experimental program is presented in which a Stanton tube is used as a direct measure of the local surface shearing stress. Calibration of the tube was made by means of a floating-element type of skin-friction balance. Calibration conditions range in Reynolds number based on free-stream conditions from 70,000 to 875,000.

Supersonic Airfoil Performance With Small Heat Addition.
Artur Mager. *Journal of the Aero/Space Sciences*, v. 26, Feb. 1959, p. 99-107.
An analytical method is presented which permits a very rapid evaluation of the aerodynamic effects arising from the addition of small amounts of heat near supersonic two-dimensional air-foils. It is shown that even small amounts of heat generate a substantial pressure rise and thus cause appreciable changes in the aerodynamic coefficients.

Effect of Uniformly Distributed Roughness on Turbulent Skin-Friction Drag at Supersonic Speeds.
Frank E. Goddard, Jr. *Journal of the Aero/Space Sciences*, v. 26, Jan. 1959, p. 1 + 15 pp.
An experimental program was carried out in the 18-in. by 20-in. supersonic wind tunnel of the Jet Propulsion Laboratory to determine the effect of uniformly distributed sand-grain roughness on the skin-friction drag of a body of revolution for the case of a turbulent

boundary layer. The Mach Number range covered was 1.98 to 4.54, and the Reynolds Number varied from about 3×10^6 to 8×10^6. Some data were also obtained at a Mach Number of 0.70. At speeds up to a Mach Number of 5 and for roughness sizes such that the quadratic resistance law holds, the compressibility effect is indirect, and the skin-friction drag is a function of only the roughness Reynolds Number, exactly as in the incompressible case. It is shown that the entire compressibility effect is a reduction of the fluid density at the surface as the Mach Number increases.

A Low-Speed Experimental Investigation of the Effect of a Sandpaper Type of Roughness on Boundary-Layer Transition.
Albert E. von Doenhoff and Elmer A. Horton. *NACA Rept. 1349*, 1958. ii, 16 pp., diagrs., photos.
An investigation conducted in the Langley low-turbulence pressure tunnel at low speeds showed that for sandpaper-type roughness the occurrence of transition could be correlated on the basis of a local roughness Reynolds number based on the size of the roughness and the velocity at the top of the roughness. The extent of the roughened area does not appear to have an important effect on the critical value of the roughness Reynolds number.

An Experimental Study at a Mach Number of 3 of the Effect of Turbulence Level and Sandpaper-type Roughness on Transition on a Flat Plate.
Robert A. Jones. *NASA Memorandum 2-9-59L*, March 1959. 43 pp., diagrs., photos., tab.
An optical method was used to study the effect of turbulence level and sandpaper-type roughness on transition for a flat plate at a Mach number of 3 with no heat transfer. A range of roughness Reynolds number and a range of roughness height to boundary-layer displacement thickness which moved transition less than 10 percent were found and were functions of turbulence level.

Boundary-Layer-Transition Measurements on Hemispheres of Various Surface Roughnesses in a Wind Tunnel at Mach Numbers from 2.48 to 3.55.
Angelo Bandettini and Walter E. Isler. *NASA Memorandum 12-25-58A*, March 1959. 29 pp., diagrs., photos., tab.
Wind-tunnel tests have been made to determine the location of transition of the boundary layer on three separate hemispheres having absolute surface roughness values of 50, 580, and 2760 microinches. The tests were in the range of Reynolds numbers, based on the 19-inch hemisphere diameter, from 1.5 to 7.5 million. Transition location was determined from shadowgraphs and checked by local temperature-recovery measurements.

3. Lubrication and Lubricants

3.1. *Lubrication, mechanism*

Effect of Lubricant on Penetration of a Diamond-Pointed Indenter Into Ductile Metals.
V. D. Kuznetsov and A. I. Loskutov. *Henry Brutcher Translation No. 4406*, 8 pp. (From *Fizika Metollov i Metallovedenie*, v. 2, no. 3, 1956, p. 509-513.)
Comparative depths of diamond indenter measurements made by authors under loads of 0 to 4 kg on dry specimens and on specimens coated with 0.2% solutions of stearic or oleic acid, respectively, in vaseline oil, of iron, copper, and zinc. Relationship between initial hardness of specimen and the load on the indenter needed to produce an effect attributable to the surface-active lubricant.

An Examination of Lubrication With Soap, Using Radioactive Sodium Stearate.
J. Golden, P. R. Lancaster, and G. R. Rowe. *International Journal of Applied Radiation and Isotopes*, v. 4, nos. 1/2, Dec. 1958, p.30-35.
Describes an experimental study of the behavior of a soap coating on rectangular mild steel bars drawn slowly under plane strain conditions between hard cylindrical dies. Loads up to 5 tons were applied to reduce the cross-section of the bars by 5-30% in one pass. Purified sodium stearate was neutron-activated and was used as the lubricant. It is found that although the drawing conditions are designed to discourage hydrodynamic lubrication, films of soap over 1000 Å thick can pass between the dies and the specimens. The soap film thickness decreases steadily as the percentage reduction of area of the specimen is increased and eventually at about 30% reaches a value of 600 Å, comparable with the maximum height of the surface asperities. At this stage the smooth surface of the specimen can be seen to be damaged and "pick-up" is found to occur.
See also: the contribution by Lancaster and Rowe on p. 428 of this issue.

Cam and Tappet Lubrication. III. Radioactive Study of Phosphorus in the EP Film.
E. H. Loeser, R. C. Wiquist, and S. B. Twiss. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 329-335.
Tightly bound films, which contain phosphorus and zinc, formed rapidly and then approached steady-state values in static and dynamic tests. The amount of film increased

with oil temperature (200° to 300°F) and with available surface area. Dynamic test conditions greatly increased the amount of film formed, indicating the marked effect of surface temperature, pressure or wear. Chemical and physical properties of the films appeared to vary with the conditions of film formation.

Lubrication of High-Speed High-Temperature Thrust Ball Bearings.
G. S. Reichenbach, J. B. Accinelli, and S. J. Beaubien. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 259-265.
Bearing rig tests in a once-through jet lubrication system with silicone fluid, mineral oil, and a synthetic ester lubricant resulted in friction failures at approximately 350, 650, and 800°F, respectively. With oil-air mist lubrication failure occurred at a much lower temperature. Demonstrates that deposits from lubricant decomposition did not play a significant role in the results and thus concludes that the small scale rig emphasized the deposit-free high temperature friction properties of lubricants.

3.2. *Lubrication, industrial*

Some Practical Notes on the Lubrication of Anti-Friction Bearings.
Scientific Lubrication (Special Extra Issue), Nov. 1958, p. 23 + 7 pages.
Design points for grease, high-temperature oil, and mist lubrication.

Centralised Lubrication and the Anti-Friction Bearing.
N. R. W. Morris. *Scientific Lubrication* (Special Extra Issue), Nov. 1958, p.55 + 2 pages.
While it is vitally important to select the correct lubricant for any given application, there is an equal necessity to ensure that it is introduced into the bearings in a clean condition, in the correct quantity, and at the prescribed intervals.

3.3. *Lubricants*

Choosing Lubricants for Heat and Cold.
Alden Crankshaw. *Machine Design*, v. 31, Feb. 5, 1959,p. 101-103.
Primary factors for consideration in selection of lubricating oils for machinery operating in temperature ranges between —65°F up to 800°F. Synthetics such as diesters are used in lubricants between —65°F and 130°F to prevent freezing and lubricant failure. At operating temperatures over 350°F, solid additives such as colloidal graphite or MoS_2 are used in suspension. Above 600°F the safest choice is colloidal graphite.

How ANP Turbine Lubricants Behave Under High-Level Radiation.
John M. Clark, Jr. and George C. Lawrason. *SAE Journal*, v. 66, Dec. 1958, p. 77-78.
Radiation can produce almost instantaneous failure of modern aircraft lubricants.

The Lubrication of Antifriction Bearings in a Nuclear Power Station.
A. J. Marles. *Scientific Lubrication* (Special Extra Issue), Nov. 1958, p. 33-37.
In equipment which is peculiar to a nuclear power station where a reactor takes the place of a furnace as the heat source the conditions peculiar to the reactor have to be considered, the main ones being the presence of radiation, high ambient temperature, compatibility with the coolant gas and other materials present, and the risk of lubricant being carried inside the reactor.

Lubricating a 5,000 h.p. Gas Turbine.
Scientific Lubrication, v. 11, Jan. 1959, p. 19-21.
Basic lubricating oil supply system consists of direct-driven 196 gal/min positive displacement main pump, a 140 gal/min positive displacement auxiliary pump, and 80 gal/min centrifugal emergency pump. Main pump has sufficient capacity to supply all lubricating and control oil requirements under gas turbine idling conditions. Auxiliary pump delivery capacity is sufficient for starting unit, and permits continuous plant operation in event of breakdown of main pump. Emergency pump can supply sufficient lubrication during plant shut-down and will ensure adequate heat dissipation from the bearings during cool down.

Lubricants in the Forming of Aluminum.
Marco Petronio. *Lubrication Engineering*, v. 14, Dec. 1958, p. 510-522.
A number of lubricants of known composition were evaluated in a laboratory slow-speed draw test by determining the forces obtained during the drawing. Shop test results are reported corroborating laboratory findings in the drawing of 2024, 5052 and 7075 aluminum alloys.

Lubrication of New Metals During Heat Treatment and Forming.
J. Grindrod. *Scientific Lubrication*, v. 11, Jan. 1959, p. 13-14.
Use of low-melting glass as extrusion lubricant for Ti alloys.

3.4. *Lubricants, greases*

Milling of Greases. I. The Mechanisms of Dispersion. II. Some Practical Benefits of Mechanical Dispersion.
Part I: K. H. Birkett. Part II: J. J. Dickason. *NLGI Spokesman*, v. 22, no. 11, Feb. 1959, p. 505-512.
Phases of manufacture and how they affect appearance and performance of lubricating

grease. Thermal and mechanical dispersions of soap particles in lubricating grease. Proper dispersion results in improved appearance, increase in yield, improved shear stability, uniformity of product, improved storage stability, decreased leakage tendencies, improved pumpability, and improved antibleed characteristics. Utilization of inorganic thickening agents.

The Flow of Lubricating Greases.
A. W. Sisko. *Industrial and Engineering Chemistry*, v. 50, Dec. 1958, p. 1789-1792.
Over a wide range of shear rates the data lie on a single smooth curve and are fitted by a new equation which can be used to calculate pressure drops in pipe flow and to explain the appearance of plug flow in greases.

Testing Anti-Friction Bearing Greases.
E. G. Ellis. *Scientific Lubrication* (Special Extra Issue), Nov. 1958, p. 14 + 3 pages.
Grease of the correct quality is a very satisfactory substitute for oil; it can be extremely economical in use, one filling serving for months or even years and the application can usually be managed with systems far less complicated than those required for a fluid.

3.5. *Lubricants, e.p., solids, gases*

Chemical Nature of Extreme Pressure Lubrication.
George Hugel. *Lubrication Engineering*, v. 14, Dec. 1958, p. 523-526.
Engine performance can be improved and seizing can be prevented by incorporating an organo-metallic complex in the lubricating oil. The complex is selected from materials which decompose at high temperatures and produce a solid mineral deposit. Various materials may be combined to produce lubricants capable of improved performance.

Investigation of Oil Additives for Boundary Lubrication of Railroad Journal Bearings.
Arnold Miller and Arthur A. Anderson. *ASLE Translations*, v. 1, no. 2, Oct. 1958, p. 273-280.
Boundary friction properties of the lubricated system of steel sliding on lead-base babbitt were investigated on freight car hot boxes. A friction pendulum was developed for determining the coefficients of boundary friction as a function of temperature with the specified metals lubricated with various lube oil additive blends. Lubricity additives which chemisorb to metal surfaces and form a close-packed layer were found to be effective antifriction agents.

Lubricating Properties of Lead-Monoxide-Base Coatings of Various Compositions at Temperatures to 1250° F.
Harold E. Sliney. *NASA Memorandum 3-2-59E*, February 1959. 22 pp., diagrs., photos., tabs.
Lead monoxide was the component that was primarily responsible for lubricating properties of various ceramic coatings studied. Other oxides in the formulations influenced such properties as softening points, adherence to metals, hardness, vitrifying tendencies, or chemical stability. Oxidation of the base metal during coating at elevated temperatures had important and often beneficial effects. Several of the coatings protected metals against adhesive wear at test temperatures from 75° to 1250°F. Friction coefficients ranged from 0.20 to 0.37 at 75°F and from 0.08 to 0.20 at 1250°F. The sliding velocity in all experiments was 430 ft./min.

Halogen-containing Gases as Boundary Lubricants for Corrosion-resistant Alloys at 1200° F.
Donald H. Buckley and Robert L. Johnson. *NASA Memorandum 2-25-59E*, March 1959. 27 pp., diagrs., photos., tabs.
The gases CF_2Cl_2, CF_2Br_2, CF_3Br, CF_3I, I_2 and SF_6 were used to lubricate nickel- and cobalt-base alloys up to temperatures of 1200°F. The use of nickel-base alloys avoided the corrosion encountered in gas lubrication of ferrous alloys above 600°F. A hemispherically tipped rider under a 1200-gram load slid on a disk rotating at a speed of 120 ft./min. Effective lubricants for a cobalt-base rider sliding on a nickel-base disk at temperatures up to 1200°F were CF_2Br_2 and CF_3Br. The use of CF_2Cl_2 with 1 percent SF_6 in the lubrication of 7-1/2 percent silicon-nickel rider sliding on a cobalt-base disk gave both low friction and wear up to 1200°F.
See also: *Ind. Eng. Chem.*, *51* (1959) 699-700.

4. Bearings and Seals

4.1. *Bearings*

On Critical Speeds Induced by Ball Bearings at Lower Rotating Speeds.
Toshio Yamamoto. *Bulletin of JSME*, v. 1, no. 3, Aug. 1958, p. 240-244.
When single-row radial ball bearings are used, several critical speeds may take place in a vertical shaft at certain rotating speeds lower than the major critical speed. When the bearing center line of both bearing pedestals is not in alignment and the shaft is not in a perfectly straight line, there is the possibility that these critical speeds occur. When self-aligning double-row ball bearings are used, these critical speeds do not occur; they occur only in the shaft supported by single-row radial ball bearings.

Current Status of Knowledge on Design Criteria for Bearings Used in High Temperature Aircraft Electrical Accessories.
P. Lewis, S. F. Murray, and A. A. Schwartz. *U.S. Air Force, Wright Air Development Center, WADC Technical Report* 58-519, Sept. 1958, 50 pp.
Fluids under consideration are silicones, silicate esters, silanes, mineral oils, polyphenyls and polyphenyl ethers, fluorinated organic compounds, esters, and polyolefins. In the bearing design, loose clearances, open curvatures, and land riding cages seem most desirable. M-50 tool steel bearings containing "S" Monel or Ni Resist Cages show successful performance. Two systems that appear most promising are grease-packed bearings and a modified "once through" system.

The Experimental Determination of Ball Bearing Cage Stress.
T. S. Crawford. Royal Aircraft Establishment (Gt. Brit.) October 1958. 8 pp., diagrs., photos., tabs. (Ask for CN-70213)
Dynamic measurements using strain gages were made to determine the effect of shaft misalignment on cage stress. Direct static measurements of ball to track friction were also made for different interference fits of the races and for varying degrees of misalignment.

Measurement of Miniature Ball Bearings: (A) A Review of Conventional Methods and (B) A New Method of Measuring Running Accuracy in the Assembled Form.
S. A. Root. Aeronautical Inspection Directorate (Gt. Brit.) June 1958. 12 pp., diagrs., photos. (Ask for CN-69099)
Since the methods of measuring track wobble, in particular, were found to be unsound and also because of the difficulties in assessing the running accuracy of a bearing from the results of separate measurements of individual features, new test apparatus has been designed. This operates on the principle that the combined effect of all the errors in a bearing will cause the rotating ring to oscillate in either or both radial and axial directions. The tests are carried out with the bearing mounted in a manner simulating operational practice under either axial or radial loads and with either the inner or outer ring rotating. The axial and radial oscillations resulting from the composite effect of errors in the bearing are picked up by axial and side acting heads and graphically recorded at high magnification.

Parallel Surface Thrust Bearing.
A. Cameron and W. L. Wood. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 254-258.
Theory for the thermal wedge thrust pad is given. Viscosity is allowed to vary in the direction of motion. Solutions are obtained by both analytical and relaxation methods for infinite and finite pads. Results are displayed nondimensionally and graphically.

Determination of Optimum Proportions for Hydrostatic Bearings.
A. M. Loeb and H. C. Rippel. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 241-247.
Effect of recess size on the pressure, flow, and power requirements of several hydrostatic bearing configurations is described in terms of dimensionless coefficients called performance factors. Performance factors are determined analytically, compared graphically with the electric analog solutions for this bearing, and found to be in excellent agreement. For a given bearing configuration, there is a particular recess size which will give minimum supply pump power.

4.2. *Seals*

The Sealing of Anti-Friction Bearings.
E. T. Jagger. *Scientific Lubrication* (Special Extra Issue), Nov. 1958, p. 39 + 6 pages.
Technological aspects of anti-friction bearings; review of current practice; classification of sealing methods; influence of working conditions and other factors; present-day tendencies; designing for synthetic rubber seals.

A Research Program on the Investigation of Seal Materials for High Temperature Application.
Raymond H. Baskey. *U.S. Air Force, Wright Air Development Center, WADC Technical Report* 58-181, June 1958, 64 pp.
The finest wear performance was attained when Kentanium K162B ran against Kentanium K162B at sliding speeds of 14,000 ft./min in an ambient temperature of 1100°F.

4.3. *Miscellaneous*

The Lead Bronze Bearings Manufactured by Means of Powder Metallurgy. (In Czech.)
Zdeněk Ministr. *Hutnické Listy*, v. 13, no. 12, Dec. 1958, p. 1132-1137.
The casting of lead bronze containing 30 to 40% Pb (an excellent antifriction metal) is very difficult because of excessive liquation of lead. The powder metallurgy opens an advantageous way to manufacture the thick walled self-lubricating bearings and also the thin-walled babbitt lined axle bearing steps of steel. The use of fine pulverized lead perfectly free of oxides, if necessary with small addition of graphite, prevents the growth of the pressed pieces and the "sweating" of lead. A method of manufacture of thick-walled self-lubricating bearings of lead bronze with a cavity for oil has been developed. The thin-walled axle bearing steps can be manufactured in two ways: either by preparing by rolling a band of pulverized lead bronze which is then bonded to a copper-coated steel plate by sintering or

the powder metal is sintered on a copper-coated steel plate and afterwards compressed by rolling. The methods of powder-metal rolling now in use and a new method of rolling between rolls one under another by means of a paper band are discussed.

PTFE Bearing Materials.
Modern Plastics, v. 36, no. 6, Feb. 1959, p. 123 + 4 pages.
Chemically inert polytetrafluorethylene compounds have low coefficients of friction, can give long service if properly designed for load and speed.

Hydrodynamic Lubrication of Ball and Socket Joints.
R. I. Tanner. *Applied Scientific Research, Sec. A, Mechanics, Heat, Chemical Engineering, Mathematical Methods*, v. 8, no. 1, 1958, p. 45-51.
As a step towards formation of design rules for ball and socket joints, a solution of the lubrication equations for an "ideal" joint under steady loading and rotation is presented. The friction for the ball is found to be higher, and the load capacity lower, than that of a plain bearing under similar conditions.

Fixtures for Testing Pin-End Columns.
A. W. Huber. *ASTM Bulletin*, no. 234, Dec. 1958, p. 41-45.
Design and the performance of column end fixtures with cylindrical bearing surfaces. These fixtures were designed for heavy loads (maximum 2,000,000 lb). and simulate a pin-end condition in one direction and fixed-end condition in the direction at right angles to the first. The fixtures can be used for axial or eccentric column tests with equal end eccentricities.

5. Wear and Wear Resistance

5.1. *Materials*

The Impact of Space Technology on Research and Development - Structures and Materials.
Richard V. Rhode and John C. Houbolt, NASA. (Presented before General Assembly, AGARD, Copenhagen, Denmark, October 20-29, 1958.) iii, 32 pp., diagrs. (Ask for NASA L-128).
A broad discussion is given of the structures and materials problems that are involved in design of space-flight vehicles. Problem areas outlined and attention is focused on those areas which require increased research activity. The point of view taken is that space technology is a natural extension of aeronautical technology and that it is the environment, mission, and configuration which set the structural problems. The nature of the space-flight environment and the general trends in construction are therefore discussed, and this is followed by a detailed consideration of the loads, structural design, and materials problems. Some specific examples illustrating the severity of the loads and structures problems are given.

Metals for Supersonic Aircraft and Missiles.
D. W. Grobecker, editor. 423 pp. 1958. American Society for Metals, Cleveland.
Proceedings of the conference, "Heat Tolerant Metals for Aerodynamic Applications", held January 28 and 29, 1957 at the University of New Mexico, Albuquerque, New Mexico, under the joint sponsorship of American Society for Metals and University of New Mexico.

Sintering of Columbium Powder.
O. P. Kolchin and N. P. Chuveleva. *Henry Brutcher Translation No. 4415*, 14 pp. (From *Tsvetnye Metally*, v. 13, no. 12, 1957, p. 67-70.) Henry Brutcher, Altadena, Calif.
Optimum conditions of sintering for ductile, easily cold-worked columbium metal. Variation of electrical resistivity, hardness, density, weight, and percentage of foreign elements in columbium powder compacts with sintering temperature.

Some Fundamental Features of Mechano-chemical Attack on Metals.
T. P. Hoar, University of Cambridge. (Presented at the AGARD Sixth Meeting of the Structures and Materials Panel, Paris, November 4-8, 1957.) Advisory Group for Aeronautical Research and Development. November 1957. iii, 5 pp. AGARD Rept. 158. (Ask for N-69063)
The behavior of oxide films under conditions leading to their chemical failure, and under fretting, fatigue, and static stress conditions, is discussed. Probably disarrayed surface zones in a metal surface lead to the formation of less effective oxide films and also, in defilmed metal, to rapid chemical attack. The consequent effects produced under fretting corrosion, corrosion-fatigue, and stress-corrosion conditions are considered. Recent experimental work is mentioned which indicates that metal that is yielding under mechanical stress is in an especially active chemical state. The implications of this work on the mechanism of stress-corrosion cracking are considered.

5.2. *Erosion and impact*

Erosion of Materials by Cavitation Attack.
D. Peckner. *Corrosion*, v. 15, 1959, p. 269-274; 7 fig., 3 tables, 19 ref.

The relationship between cavitation attack and erosion of metals is discussed. Extensive references are made to existing literature on cavitation attack. Data are reported on the effect of hardness on accelerated cavitation testing of chromium stainless steels, effect of sodium dichromate inhibitors on resistance of metals to pitting, and determination of the relative erosion resistance of alloys by the vibratory method. Streamlining, good surface finish and the use of cavitation resistant materials tended to reduce the cavitation effect. The relative resistance of metals to cavitation could be evaluated successfully with a two-hour magnetistrictive test.

Effect of Grain Size on the Resistance of Steels to Cavitation Erosion.
I. N. Bogachev and R. I. Mints. *Henry Brutcher Translation No. 4317*, 6 pp. (From *Metallovedenie i Obrabotka Metallov*, no. 8, Aug., 1958, p. 26-29.) Henry Brutcher, Altadena, Calif.
Relationships in single-phase (austenitic, ferritic) steels. Mechanism of cavitational destruction. Cavitation erosion in steels with duplex structures as function of grain size, component making up the network proper, and brittleness or toughness of steel; mechanism of cavitational failure.

Erosion Wear and Wear Protection of Austenitic Steels for Gas Turbine Blades (in Russian)
V. I. Prosvirin and A. I. Fedosov. *Inzhenerno-Fizicheskiĭ Zhurnal*, v. 2, no. 1, Jan. 1959, p. 6-14.
The best method of protection from erosive wear is by simultaneous saturation by C, Cr, and N, or the two latter elements, when the coefficient of resistance to erosion is increased by 10 or more. The character and mechanism of erosive wear on blades is described.

A Correlation between Rain Erosion of Perspex Specimens in Flight and on a Ground Rig.
T. J. Methven and B. Fairhead. Royal Aircraft Establishment (Gt. Brit.) November 1958. 11 pp. diagrs., photos., tabs. (Ask for CN-71206)
The amount of surface erosion on Perspex has been measured for specimens flown on an aircraft in rain and tested on a whirling arm ground rig in artificial rain. Specimens were compared at 400 knots and similar rain concentrations. Results show that 1 in./h rain in flight gives similar erosion to 1.5 in./h on the ground rig; this may be due to the greater range of droplet sizes found in flight.

Note on a Laboratory Apparatus to Study the High Speed Impact between a Liquid Drop and a Surface.
D. C. Jenkins, J. D. Booker, and J. W. Sweed. Royal Aircraft Establishment (Gt. Brit.) February 1958. 10 pp., diagrs., photos., tab. (Ask for N-65562)
An apparatus is described which is being used for the study of the effects of high-speed impact between a liquid drop and a surface. The apparatus has been used at speeds up to 1750 ft./sec. Some other applications of the apparatus are discussed.

A Preliminary Investigation of High-speed Impact: the Penetration of Small Spheres into Thick Copper Targets.
A. C. Charters and G. S. Locke, Jr. *NACA RM A58B26*, May 1958. 21 pp., diagrs., photos.
Small metal spheres of various densities were fired at high speed into thick targets of copper and lead. In general, it was found that all of the penetrations could be correlated quite well for engineering purposes by a function relating the depth of penetration to the impact momentum per unit volume.

5.3. *Miscellaneous*

Reducing Scuffing and Wear of Ferrous Metals. Surface Treatment by Sulfinuz Process.
F. D. Waterfall. *Engineering*, v. 187, Jan. 23, 1959, p. 116-120.
This process is to impregnate steel or cast iron surfaces with S and N, to reduce scuffing and wear. Surfaces are treated in a molten salt bath of cyanide and S compounds at about 570°C from 5 minutes to 3 hours. After two hours at 570°C, a "white" microscopic surface layer about 0.0005 in. in depth can be observed. Nitrogen penetrates beyond this white layer and confers some benefit to the metal below.

Severe Wear of Pearlitic Steel Weld Metal.
Sukemitu Ito, Kozo Honda, and Kikuzo Ishiyama. *Journal of Mechanical Laboratory of Japan*, v. 4, no. 1, 1958, p. 15-21.
The value indicating wear was found for 39 different brands of hardfacing electrodes. The relation between the wear property and microstructure or chemical composition was complex.

The Acceleration of Pitting Failure by Water in the Lubricant.
L. Grunberg and D. Scott. *Institute of Petroleum, Journal*, v. 44, Nov. 1958, p. 406-410.
The effect of water content of the lubricant on the incidence of pitting type failure of ball bearings was studied using the four ball rolling test. Several oils of different characteristics which are being used for the lubrication of ball bearings in practice were tested under four conditions of water content. The presence of water in the lubricant accelerated the pitting failure of balls made of conventional EN. 31 steel, but had no effect on the pitting failure of stainless steel balls.

6. Analysis and Testing

Measuring the Attack of Gases on Solid Surfaces.
Corrosion Technology, v. 6, Jan. 1959, p. 9.
Procedure employs time-lapse photography of the reacting solid so that a permanent record of the reaction's progress is obtained.

Investigation of Wear by Electron Diffraction.
L. M. Niebylski, M. Antler, and L. O. Brockway. *ASLE Transactions*, v. 1, no. 2, Oct. 1958, p. 304-311.
Discusses monolayers, liquid lubricants, and greases, polished and amorphous films, the relation to wear of the crystallite size of surface films, solid lubricants, mechanism of extreme pressure lubrication, and the identification of wear debris. Molybdenum isobutylxanthate is an effective extreme pressure oil additive for hard steel. Cast iron top-piston-ring durability in a spark-ignition engine was studied using several fuels which gave widely different wear rates.

Recherche d'un test rapide d'abrasivité.
An Attempt to Find a Quick Abrasivity Test. P. Draignaud. *Société Francaise de Céramique, Bulletin*, no. 41, Oct.-Dec. 1958, p. 47-64.
Test involves rotating the metal bar to be tested in a bed of quartz grains of 10-15 mm size.

Contribution à l'étude de l'usure d'un moteur par la technique des traceurs radioactifs.
Study of motor wear by tracer technique.
R. Wilputte. *Belg. Chem. Ind.*, v. 12, 1958, p. 1374-1378; 3 fig., 6 ref.
A simple apparatus is described for the measurement of engine wear by the radioactive ring method. This process has proved useful: for comparing the "anti-wear" properties of different lubricants during continuous run and on starting; for investigating how engine wear on starting can be reduced.

Preliminary Study of A Piston Pump for Cryogenic Fluids.
Arnold E. Biermann and Robert C. Kohl. *NASA Memorandum 3-6-59E*, March 1959. 27 pp., diagrs., photo's.
Preliminary performance data are presented for a low-speed, five-cylinder piston pump designed to handle boiling hydrogen. The pump is designed to deliver 55 gallons per minute at a discharge pressure of 135 pounds per square inch. The design pump speed is 240 rpm. The report also describes associated cavitation experiments in which liquid nitrogen and water were subcooled at 0.6° F by reciprocating a disk in an open vessel containing the test fluids. This method produced a lesser amount of subcooling with liquid hydrogen.

7. Surface Treatment and Finishing

Effect of Surface on the Behaviour of Metals.
100 pp. + 10 plates. 1958. Philosophical Library, New York.
Contains the four papers presented at the Institution of Metallurgists' annual Refresher Course for 1957, entitled "Methods of Preparation and Examination of Surfaces"; "Influence of Surface Treatments on the Chemical Behaviour of Metals"; "Relationship Between Surface Condition, Friction and Wear"; and "Influence of Surface on the Physical Properties of Metals".

The Nature of Mechanically Polished Metal Surfaces: An Electron-Diffraction Examination of Polished Silver Surfaces.
L. E. Samuels and J. V. Sanders. *Institute of Metals, Journal*, v. 87, pt. 5, Jan. 1959, p. 129-135.
Electron-diffraction patterns characteristic of crystalline material, obtained from surfaces of both polycrystalline and single-crystal silver polished mechanically by a standard metallographic method, show that the range of the misorientations—and hence the magnitude of the deformation at the surface—produced by polishing decreases progressively with increasing fineness of polish. With the finest polish investigated, the misorientations were only about $\pm 5°$. No evidence was found of a layer of peculiar structure of the type postulated by Beilby.

Finishing and Inspection of Model Surfaces for Boundary-layer-transition Tests.
Max E. Wilkins and John F. Darsow. *NASA Memorandum 1-19-59A*, February 1959. 17 pp., diagr., photos.
Techniques which have been used for finishing and quantitatively specifying surface roughness on boundary-layer-transition models are reviewed. The appearance of a surface as far as roughness is concerned can be misleading when viewed either by the eye or with the aid of a microscope. The multiple-beam interferometer and the wire shadow method provide the best simple means of obtaining quantitative measurements.

8. Machining and Tool Wear

8.1. *Grinding*

New Techniques for Grinding Sintered Carbides.
E. Zmihorski. *Machinery (London)*, v. 93, Nov. 19, 1958, p. 1186-1188.
New method of rough grinding involves raising the temperature of the part to be ground to within 750 to 850°C, and maintaining it at this temperature during grinding. It has been found that at this temperature the sintered carbides are considerably less brittle, and, consequently, less prone to craze cracking.

Basic Tests of Grinding Wheel Hardness.
P. Roy Leaman. *Tooling & Production*, v. 24, no. 12, Mar. 1959, p. 63-64.
Describes testing by use of tone or sound, density or penetration methods.

The Influence of Different Factors on Metal Removal and Abrasive Consumption in Grinding Steel With Pendulum Grinding Machines. (in Swedish)
Gunnar Wallquist, Bo Hamrin, and Hans Lund. *Jernkontorets Annalar*, v. 152, no. 9, 1958, p. 541-569.
The influence of different factors on the metal removal and the consumption of abrasives during grinding of rolled steel billets with pendulum grinding machines were determined at several Swedish steel works. Nineteen grinding wheels of different types were tested, and of these five were investigated in detail at grinding pressures of 80, 100, and 120 kg and at wheel periphery speeds of 35, 40, 45, and 50 m/sec.

8.2. *Drilling*

Hard Metal—The New Rock Drilling Medium.
L. S. Cole. *Australasian Engineer*, 1959, Jan. 7, p. 70 + 2 pages.
In percussion drilling, the average drilling "life" of a forged steel or detachable head is about 42 inches. A hard metal tip can be used at greater drilling speeds and has an average life of about 35 feet before re-sharpening is necessary. Describes manufacture and use of hard-metal tips.

Vibration Drilling Gets Further Study.
Ralph Simon. *Drilling*, v. 20, Feb. 1959, p. 110.
Drilling by vibration is accomplished by superimposing longitudinal vibrations onto the drill bit of an otherwise conventional rotary drilling system. A number of laboratory and field models of vibratory drilling machines have been designed, constructed, and used experimentally. The latest version encompasses a magnetostriction transducer that follows a 10½-in. rotary bit down the hole. It is tentatively concluded that vibratory drilling in rock of medium hardness may be just about economically competitive with conventional drilling.

8.3. *Machining*

Equipment Requirements for High-Speed Machining.
H. J. Siekmann. *Tool Engineer*, v. 42, no. 1, Jan. 1959, p. 49-52.
Ceramic tools can cut four times as fast as carbides. In order to utilize this potential productivity, machine tools of new design are required. Characteristics of such machines are outlined.

Electro-Erosion Machining of Cemented Carbides. (in Czech)
V. Vícha. *Strojírenská Výroba*, v. 7, no. 1, 1959, p. 11-14.
Basic data about the method and examples of the process of machining various parts.

Hard Spots in the Machined Surface.
Antoni Niedzwiedski. *Tooling & Production*, v. 24, no. 12, Mar. 1959, p. 57-59.
Tries to explain how, while machining with feeds too fine, the cutting edges of milling cutters may slide over instead of cut the machined metal. These sliding cutting edges cold work the machined surface and thus form the hard spots. The very common phenomenon in finish milling steel with HSS milling cutters, known usually under the shop name of "hard spots", might not be due to any property of the machined metal, but simply to certain inappropriate working conditions.

8.4. *Tool wear*

A Photo-Elastic Analysis of Machining Stresses on Rake Face.
Eiji Usui and Hidehiko Takeyama. *Journal of Mechanical Laboratory of Japan*, v. 4, no. 1, 1958, p. 6-14.
The stress distributions on the rake face were successfully determined photo-elastically by using a tool made of epoxy resin. The frictional stress on the rake face is uniform over a wide range of the tool-chip contact area regardless of the varying normal stress thereon, but it decreases steeply near the point of chip-separation.

Friction on Relief Face of Cutting Tool.
M. Okoshi and T. Sata. *J. Sci. Research Inst. (Tokyo)*, v. 52, Dec. 1958; 7 fig., 3 ref.
Friction on the relief face of cutting tools is discussed in comparison with that on the rake face. Two components of the cutting force, shear angle and contact length on the

rake face with chip are measured for different lengths of contact between the relief and machined surfaces. It is found that the compressive stress on the relief face remarkably increases with the increase in contact length on the relief face, and also that the shear stress on the relief face remains unaltered by the changes in contact length, feed of tool and relief angle. Further, the shear stress on the relief face is found about equal to that on the rake face. From these results, it is shown that the friction on the relief face can be satisfactorily explained as due to metal-shearing such as observed on the rake face.

Neuere Untersuchungen über den Zusammenhang zwischen der Spanentstehung und dem Standzeitverhalten beim Drehen von Stahl mit Hartmetall-Legierungen.
Recent Investigations on the Relation Between the Formation of the Chip and Tool Life when Turning Steel with Carbide Tips.
Günter Weber. *Stahl und Eisen*, v. 78, no. 23, Nov. 13, 1958, p. 1678-1690.
Relations between characteristic factors governing chip formation: shear stress within the shear zone, chip thickness ratio, main cutting force, velocity of the chip during sliding over the tool face, width of the contact area between chip and tool face and average pressure on the tool face.

Evaluating Tool Life.
D. Peckner and H. Ginsburg. *Tool Engineer*, v. 42, no. 3, Mar. 1959, p. 75-79.
Statistical methods when applied to tool wear testing can predict performance quickly. Describes a useful technique and illustrates the principles with a report of a study performed on form-cutters used to machine slots.

Book Reviews

Mechanical Properties of Non-Metallic Brittle Materials, Edited by W. H. WALTON, Butterworths Scientific Publications, London; Interscience Publ., Inc., New York, 1958). 25 × 17 cm; xi + 492 pp.; numerous figures and tables; price £ 4.10.0, ($ 12.75)

Proceedings of a Conference on Non-Metallic Brittle Materials organized by the Mining Research Establishment of the National Coal Board in consultation with the Building Research Station (D.S.I.R.) and held in London, April 1958.

The brittle behaviour of metals and polymers has been the subject of many fundamental papers, but industry is also interested in the mechanical properties of such complex materials as coal, bricks, concrete and rocks, which are heterogeneous even on a macroscopic scale and therefore difficult to study. The British National Coal Board took the initiative in organizing a conference at which papers were read on work being carried out on coal at their Mining Research Establishment, together with contributions from university and industrial laboratories dealing with related topics.

These Proceedings, containing the contributions as well as the interesting discussions that followed, can be warmly recommended to engineers, physicists and chemists who have a professional interest in the mechanical properties of solids. There were four Sessions:

I. *Strength in Compression, Tension, Bending and Shear* (153 pp.) 8 Papers dealing with 6 widely different brittle materials.
II. *Elasticity and Creep* (125 pp.)
7 Papers on the dynamic properties of concrete, ceramics, coals and graphite.
III. *Dynamic Loading, Impact and Fragmentation* (115 pp.)
7 Papers on concrete, coal and rocks, their strength properties and the size distribution of fragments.
IV. *Action of Tools* (85 pp.)
5 Contributions from the N.C.B. on problems related to coal ploughing, with a contribution on "friction between coal and metal surfaces". Other papers deal with the cutting processes in ploughing coal and certain similarities with the cutting of metals.

A book of this type does not make easy reading. Most of the contributions refer to work still in progress and confront the reader with a mass of data and technical details. This, however, should make the work of struggling through the contents worthwhile.

New applications of dynamic testing, new experimental approaches and accurate data, some reported for the first time, are of immediate practical importance in the mechanical technology of these brittle solids.

The volume is well produced and should be a valuable addition to the Library, even of an Institute concerned only with metals or polymers.

G.Sa.

Hydro-Aerodynamics of Lubrication (*Hydro-Aerodinamica Lubricatiei*) (in Rumanian), by N. TIPEI, Biblioteca Stiintelor Tehnice, Editura Academici Republicii Populare Romine, Bucuresti, 1957, Vol. 1, 24 × 18 cm; 695 pp.; about 100 fig.

The contents of this book, published in Rumanian, are fairly easy to grasp — even without a dictionary — for the reader with a knowledge of Latin. A mimeographed English translation of the pages of contents is supplied with the book; moreover, the essential parts are mathematical deductions, tables and figures, familiar to the specialist.

The aim of the author is to give a detailed picture of the theory of hydrodynamic lubrication and its application to engineering problems. The thorough theoretical treatment is up to date in its references to world literature. A greater variety of problems seems to be treated than in most other texts; solutions of a number of new problems are also included. The numerical examples are taken from Professor Tipei's own researches as well as from other sources. Only in a few cases is theory proved by published experimental work, performed in other laboratories.

The first three chapters give an introduction to friction and lubricants followed by an exposition of the theory of slider bearings (84 pages). The bulk of the book is formed by the following chapters:

IV. Bearings with constant forces and velocities (243 pages).
V. Bearings with no radial clearance (30 pages).
VI. Bearings with variable geometric elements (90 pages).
VII. Bearings with variable forces and velocities (70 pages).
VIII. Stability of motion of lubricated bodies (30 pages).
IX. A more general study of hydrodynamic lubrication (70 pages).
X. Gas lubrication (65 pages).

This book is written for physicists and engineers who wish to specialize in the field of lubrication. Numerous cross references, clear printing of the mathematical symbols and welldrawn illustrations will assist the reader in finding points of special interest.

H.B.
G. Sa.

Notes on Contributors

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J. F. ARCHARD: (for biographical note see *Wear*, 2 (1958/59) 84). [See p. 438]

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J. F. OSTERLE: (for biographical note see *Wear*, 1 (1957/58) 530). [See p. 416]

G. W. ROWE: (for biographical note see *Wear*, 1 (1957/58) 530). [See p. 428]

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